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Proceedings of the 1st Symposium on Advances in Refrigeration and Heat Pump Technology

Elmegaard, Brian; Brix, Wiebke; Kærn, Martin Ryhl; Ommen, Torben Schmidt; Wronski, Jorrit; Holten-Tingleff, Frederik; Poulsen, Claus S.; Jensen, Jørn Borup ; Skovrup, Morten Juel; Jakobsen, Arne

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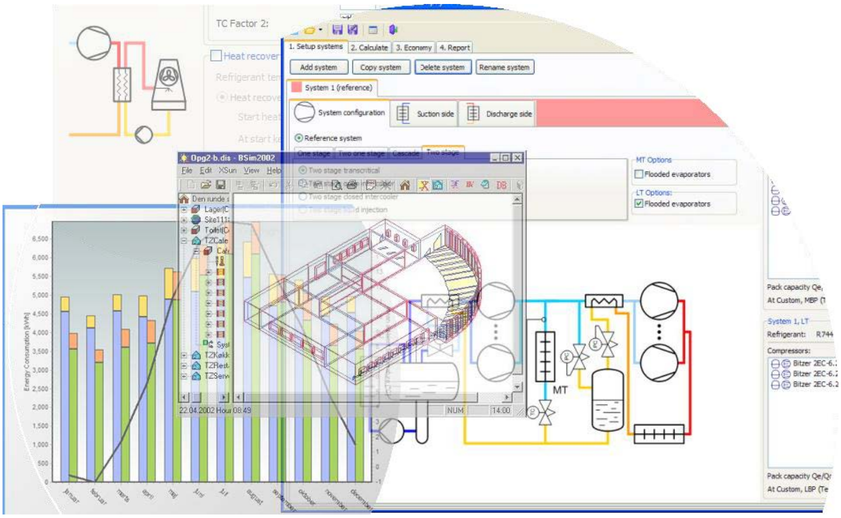
1st Symposium on Advances in Refrigeration and Heat Pump Technology

15–16th May 2012, Kongens Lyngby, Denmark

Proceedings



Edited by

Brian Elmegaard, Wiebke Brix, Martin Ryhl Kærn, Torben Ommen, Jorrit Wronski, Frederik Holten-Tingleff, Claus S. Poulsen, Jørn Borup Jensen, Morten Juel Skovrup, Arne Jakobsen

May 2012

DANISH TECHNOLOGICAL INSTITUTE

DTU Mechanical Engineering
Department of Mechanical Engineering

DANISH ENERGY ASSOCIATION

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1 Preamble

1.1 Foreword

Technical University of Denmark – Department of Mechanical Engineering, Danish Technological Institute, and the Danish Energy Association in collaboration hosted a two-day symposium covering advances in refrigeration and heat pump technology on the 15th and 16th of May 2012.

These proceedings are the formal documentation of the lectures that were given over the two days on several topics of significant relevance for the future development of technology for cooling and heating application. The focus was on both the industrial development of solutions for domestic, commercial and industrial applications in the near future as well as the scientific and engineering research in the more distant years to come. Applications of compression technology, phase changing materials and magnetic refrigeration were presented as well as novel results for selection of working fluids and design of cycles, development of components and cycles. The topics were presented by experts in the specific fields who had taken the opportunity to disseminate the results of their most recent research to an audience covering practitioners in the fields of refrigeration and heat pumps together with consultants, development engineers and staff from academia who together formed a forum for fruitful discussions at high-level in an open and responsive atmosphere.

We had invited a number of keynote speakers to give talks on topics which were intended to give inputs on the position of research and development in refrigeration technology in the future energy system:

- Hans Hvidtfeldt Larsen, DTU National Laboratory for Sustainable Energy, DTU International Energy Report
- Anders Stouge, Danish Energy Association, Research and Development in Efficient Energy Use
- Henrik Lund, Aalborg University, Heat Pump Integration in Energy Systems

And on recent development trends with significant commercial perspectives in Denmark and the neighbouring countries Sweden and Germany:

- Per Henrik Pedersen, DTI, Center for Refrigiation, Energy Efficient Impulse Coolers
- Michael Kauffeld, Karlsruhe University of Applied Sciences, Germany, Minichannel Heat Exchangers

- Hatef Madani, Royal Insitute Of Technology, KTH, Sweden, Capacity-Controlled Ground Source Heat Pumps

The keynote talks inspired further discussion and we hope that they may have sowed just a small seed of an idea which may be harvested in future R&D efforts.

We would like to thank the speakers and the participants for their mindful attendance and active participation. It was a pleasure to host the event. We hope this has been the initial meeting in a range of symposia in the field and we are of the definitive persuasion to take part in arranging future meetings. We are very open for topic suggestions for future events.

Wiebke Brix, DTU Mechanical Engineering
Brian Elmegaard, DTU Mechanical Engineering
Frederik Holten-Tingleff, DTU Mechanical Engineering
Arne Jakobsen, Københavns Maskinmesterskole
Jørn Borup Jensen, Danish Energy Association
Martin Ryhl Kærn, DTU Mechanical Engineering
Torben Ommen, DTU Mechanical Engineering
Claus S. Poulsen, DTI, Energy & Climate
Morten Juel Skovrup, IPU Refrigeration and Energy Technology
Jorrit Wronski, DTU Mechanical Engineering

1.2 Timetable

Tuesday 15th of May, Mødelokale 1		
9:00	Brian Elmegaard	Welcome
9:10	Hans Hvidtfeldt Larsen	DTU International Energy Report
10:00	Hatef Madani	Capacity-Controlled Ground Source Heat Pumps
11:10	Anders Stouge	Research and Development in Efficient Energy Use
11:45	Wiebke Brix	Second-law Efficiency and COP of Supermarked Refrigeration Systems
12:15		**Lunch Break**
13:15	Svend V. Pedersen	Application of Industrial Heat Pumps
14:00	Lars Reinholdt	Refrigerant Flow in Vertical Pipes
14:45		**Short Break**
15:00	Martin Ryhl Kærn	Compensation of Airflow Maldistribution in Fin-and-Tube Evaporators
15:45	Peter Schneider	Cooling Towers of the Future
16:30		**Short Break**
16:45	Michael Kauffeld	Minichannel Heat Exchangers

Tuesday 15th of May, Mødelokale S09		
13:15	Mikael Bastholm	Aquifer Thermal Energy Storage
14:00	Jorrit Wronski	Pumpable Phase Change Material
14:45		**Short Break**
15:00	Christian Bahl	Magnetic Refrigeration – and Heating
15:45	Ulrik Larsen	Engineering the Heat Curve

Wednesday 16th of May, Mødelokale 1		
9:00	Per Henrik Pedersen	Energy Efficient Impulse Coolers
9:45	Christian Heerup	Optimization of Commercial Refrigeration Plants
10:30		**Short Break**
11:00	Henrik Lund	Heat Pump Integration in Energy Systems
11:45	Morten Juel Skovrup	Optimization Through Interaction Between PackCalc and BSim
12:30		**Lunch Break**
13:30	Torben Ommen	Heat Pump Booster Configurations in Novel District Heating Networks
14:15	Gunda Mader	Cost and Energy Efficiency of Air-Water Heat Pumps
15:00		**Closing Plenum Discussion**

1.3 Welcome

Brian Elmegaard *(be@mek.dtu.dk)*
DTU Mechanical Engineering

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Symposium on Advances in Refrigeration and Heat Pump Technology

15-16 May 2012

- Symposium: A Hellenic drinking party
- It was a forum for men of good family to debate, plot, boast, or simply to revel with others
- A "symposiarch" would decide how strong the wine for the evening would be, depending on the seriousness of the discussions

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Symposium on Advances in Refrigeration and Heat Pump Technology

15-16 May 2012

A forum
- for dissemination of recent research results
- for being informed about recent results

- Sufficient time for presentations
- Sufficient time discussions

- Keynote talks by
  - Hans Larsen, DTU
  - Anders Stouge, Dansk Energi
  - Henrik Lund, AAU
  - Hatef Madani, KTH
  - Per Henrik Pedersen, DTI
  - Michael Kauffeld, Karlsruhe

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Symposium on Advances in Refrigeration and Heat Pump Technology

15-16 May 2012

Organizing committee

Brian Elmegaard, DTU
Wiebke Brix, DTU
Martin Kærn, DTU
Torben Ommen, DTU
Jorrit Wronski, DTU
Frederik Holten-Tingleff, DTU

Claus Schøn Poulsen,
Danish Technological Institute

Jørn Borup Jensen,
Danish Energy Association

Morten Skovrup, IPU

Arne Jakobsen, KME

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2 Keynotes

2.1 Minichannel Heat Exchangers

Michael Kauffeld *(michael.kauffeld@hs-karlsruhe.de)*
Karlsruhe University of Applied Sciences, Germany

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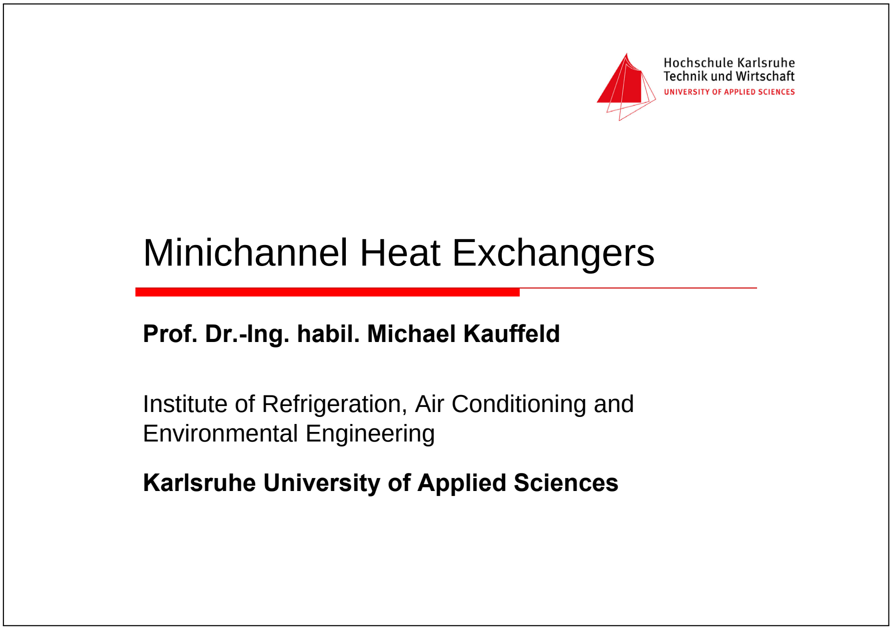
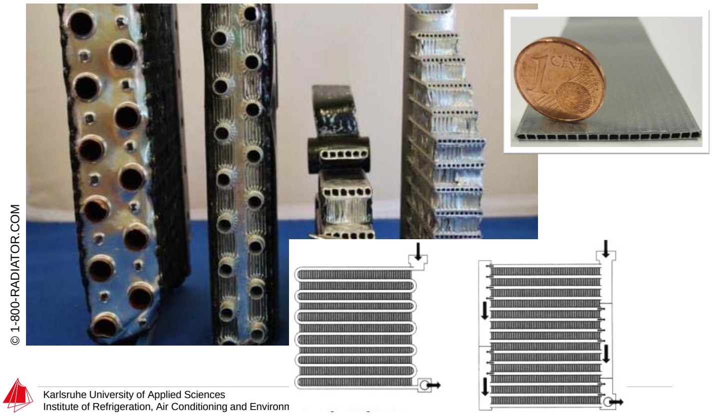


Heat Exchanger Development (automotive AC condensers)

Large diameter round tubes	Small diameter round tubes	Serpentine MPE	Parallel Flow Minichannel



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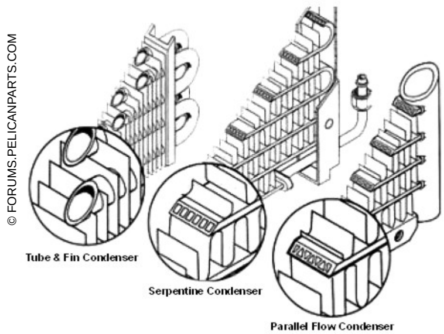
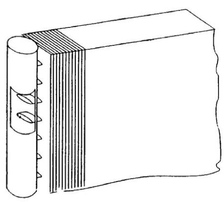
Heat Exchanger Development





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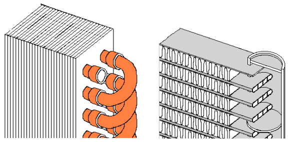
Advantages Minichannel Condensers



	Round tube and fin	Minichannel
Depth	100 %	28 %
Face area	100 %	75 %
Weight	100 %	42 %
Refrigerant charge – in Condenser in System	100 % 100 %	7 % 65 - 70 %
Air side pressure drop	100 %	74 %
COP	100 %	110 %

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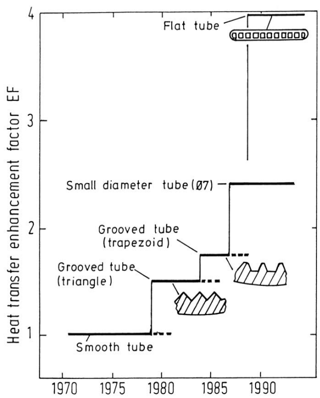
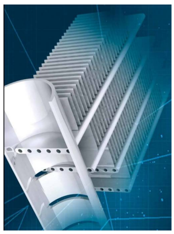
… at the cost of higher refrigerant side pressure drop

Minichannel Heat Exchanger





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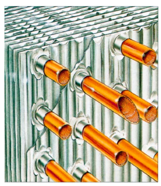

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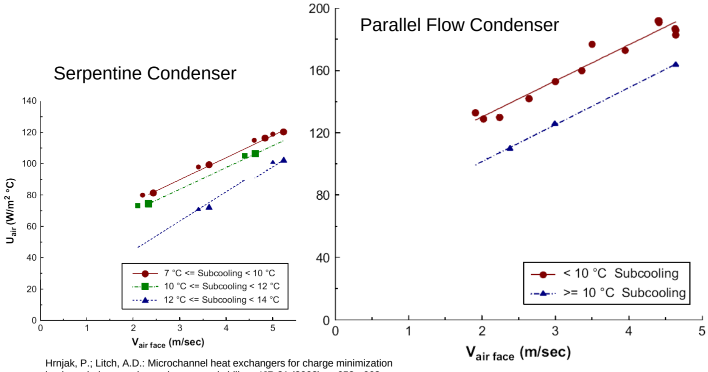
Marvillet, C.: Recent Developments in Heat Exchangers for Automotive Applications. in Recent Development in Finned Tube Heat Exchangers. Edited by Marvillet, C. et al., 1990, Danish Technological Institute, Denmark, S. 8 - 51

Comparison Serpentine vs. Parallel Flow Condenser

Overall heat transfer coefficient based on air side



Hrnjak, P.; Litch, A.D.: Microchannel heat exchangers for charge minimization in air-cooled ammonia condensers and chillers. IJR 31 (2008), p. 658 - 668

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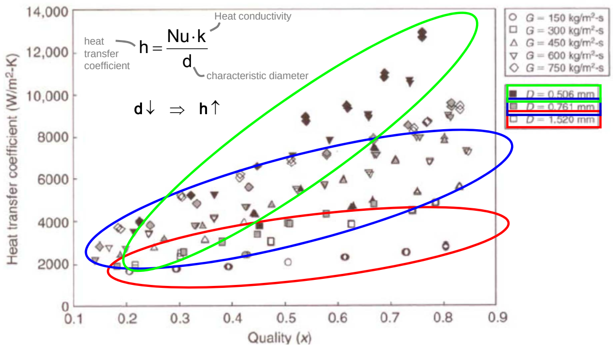
Small Channel Diameters improve Heat Transfer



Bandauer, T.M.; Agarwall, A.; Garimella, S.: Measurement and modelling of con-densation heat transfer in circular microchannels. Journal of Heat Transfer (2006)

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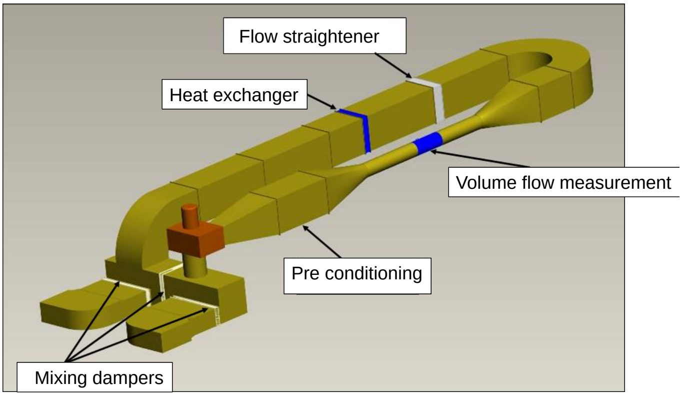
Heat Exchanger Wind Tunnel in Karlsruhe

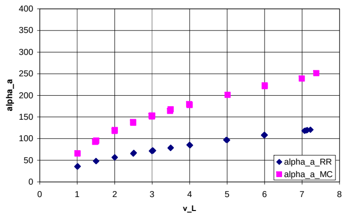
Overall heat transfer coefficient based on air side



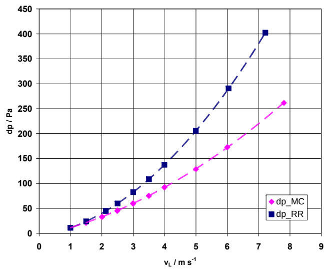
Comparison Round Tube and Fin vs. Minichannel

$\dot{Q}c$ = 10 kW
Face area 600 x 600 mm
Depth 95 mm bzw. 20 mm

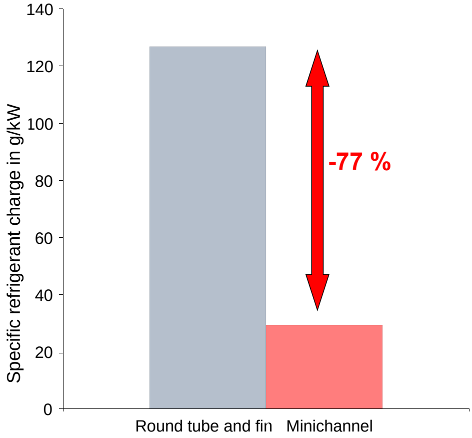
Air side heat transfer coefficient



Air side pressure drop



Refrigerant Charge



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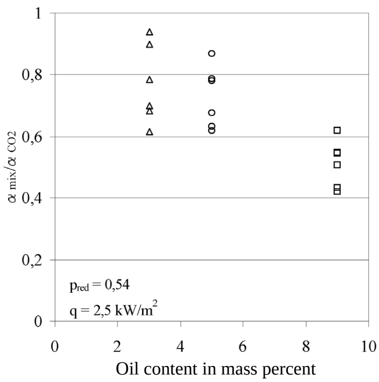
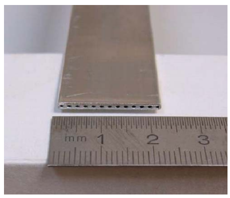
But: Oil Influence on Heat Transfer





Siegismund, V.; Kauffeld, M.: Influence of Lubricant Oil on CO_2 Heat Transfer in Minichannel Tubes. IIR Conference Thermophysical Properties and Transfer Processes of Refrigerants, Vicenza, Italy, 30.8. – 2.9. 2005

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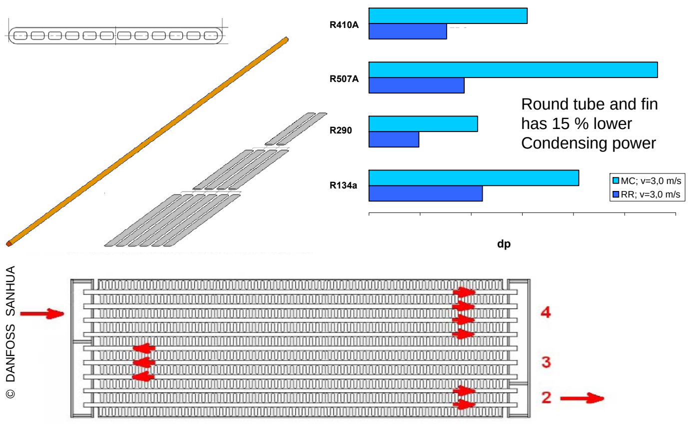
Refrigerant Side Pressure Drop



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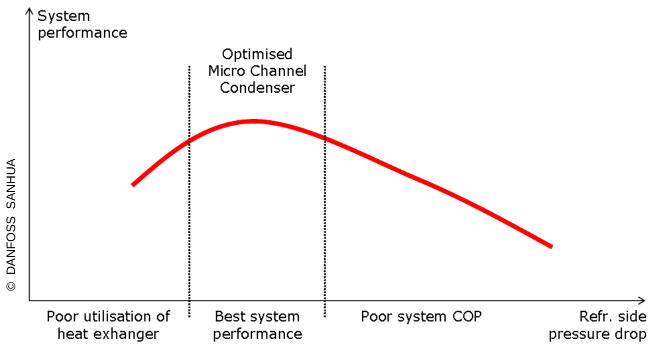
Refrigerant Side Pressure Drop



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Dirt Accumulation

3 row f&t HX after 24 hours

20 mm MCHX after 24 hours

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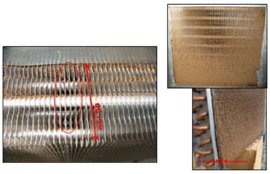
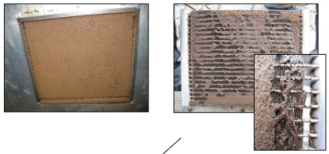
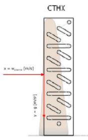
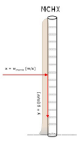




f&t gets dirty and loose capacity

Difficult to remove dirt efficiently and without damaging the fins





MCHX get dirty and loose capacity

Dirt accumulates on front and can be removed easy and efficiently without damaging the fins

Micro Channel Heat Exchang

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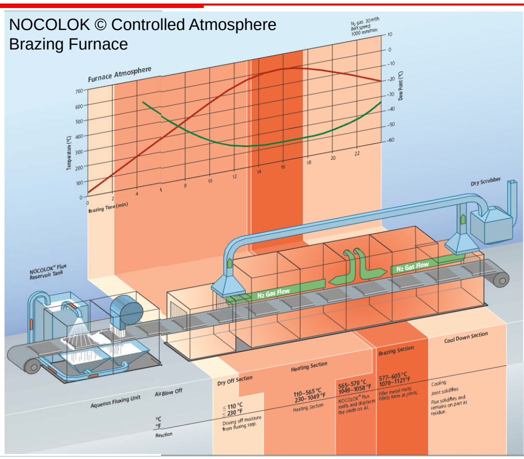
Manufacturing requires different technology



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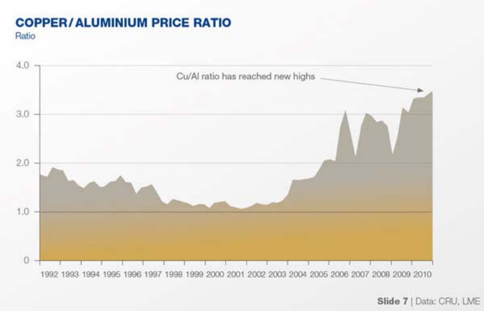
Metal Price in favor for Minichannel Heat Exchangers



© ICF.AT; data from CRU and London Metal Exchange

In addition density of alumium 2.7 g/cm^{-3} copper 8.9 g/cm^{-3} i.e. a factor of 3
→ per volume copper is more than 10-times more expensive !

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Outlook for Minichannel Heat Exchangers

- ☐ Refrigerant distribution
- ☐ Minichannel profile geometry
- ☐ Header tubes

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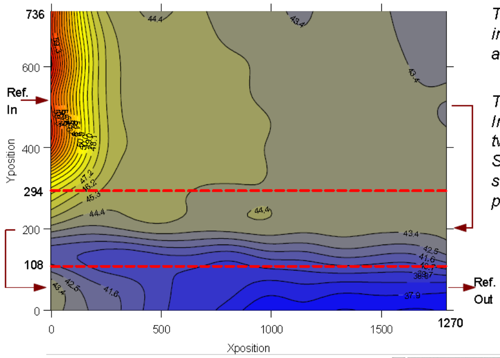
Refrigerant Distribution



The exit air temperature indicates refrigerant distribution and refrigerant temperature.

The condenser has 3 passes. In first pass, superheated and two-phase regions exist. Stratification in second and the subcooled region in the third pass is evident

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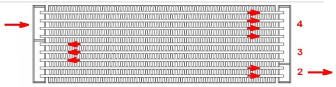

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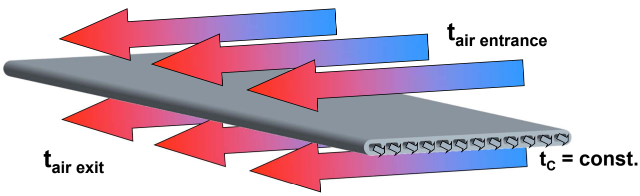
Maldistribution in MPE - Condenser



$t_{air\ exir} > t_{air\ entrance}$ → $\Delta t_{entrance} > \Delta t_{exit}$

and thus: $\dot{Q}_{entrance} > \dot{Q}_{exit}$

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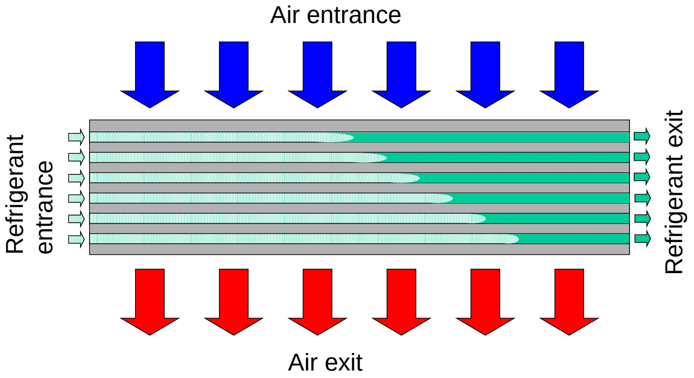
Maldistribution in MPE - Condenser



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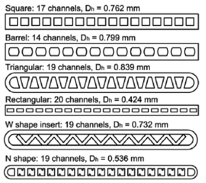
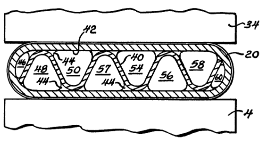
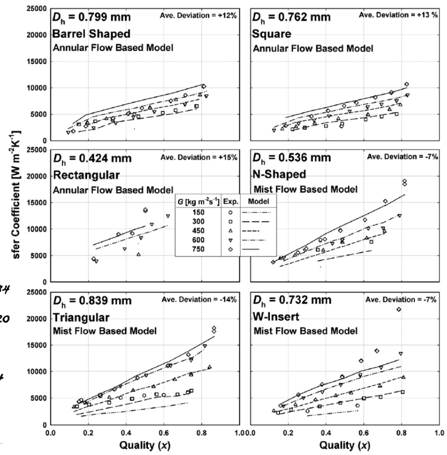
Designated condensate drainage



Agarwal, A.; Bandhauer, T.M.; Garimelly, S.: Measurement and modelling of condensation heat transfer in non-circular microchannels. IJR 33 (2010) p. 1169 - 1179



EP0219974: Condenser with small hydraulic diameter flow path. Filed 17.09.1986 by MODINE



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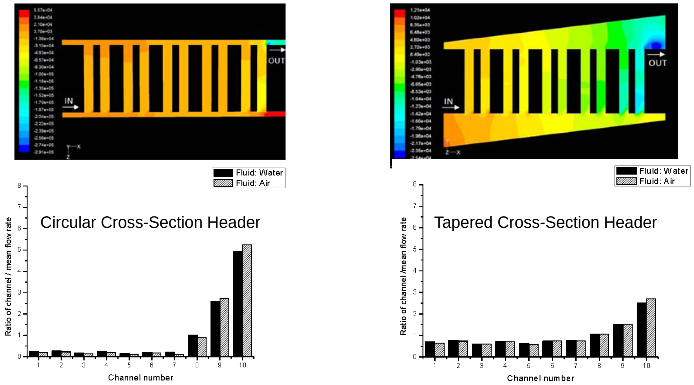
Optimized Distribution - Evaporator



Dharaiya, V.V.; Radhakrishnan, A.; Kandlikar, S.G.: Evaluation of a tapered header configuration to reduce flow maldistribution in minichannels and microchannels. Proceedings of the ASME 2009 7th International Conference on Nanochannels, Microchannels and Minichannels, June 22-24, 2009, Pohang, South Korea

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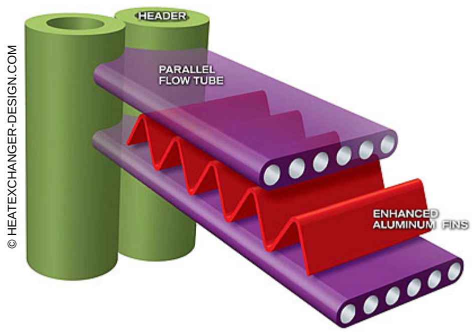
Minichannel HX for high pressure (R744)

Special header design for high pressure + round holes in MPE

… could also be used for further charge reduction



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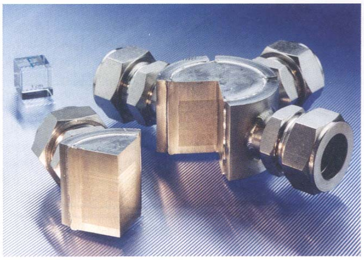
Minichannel – the End of Miniaturisation ?

Cross flow micro heat exchanger



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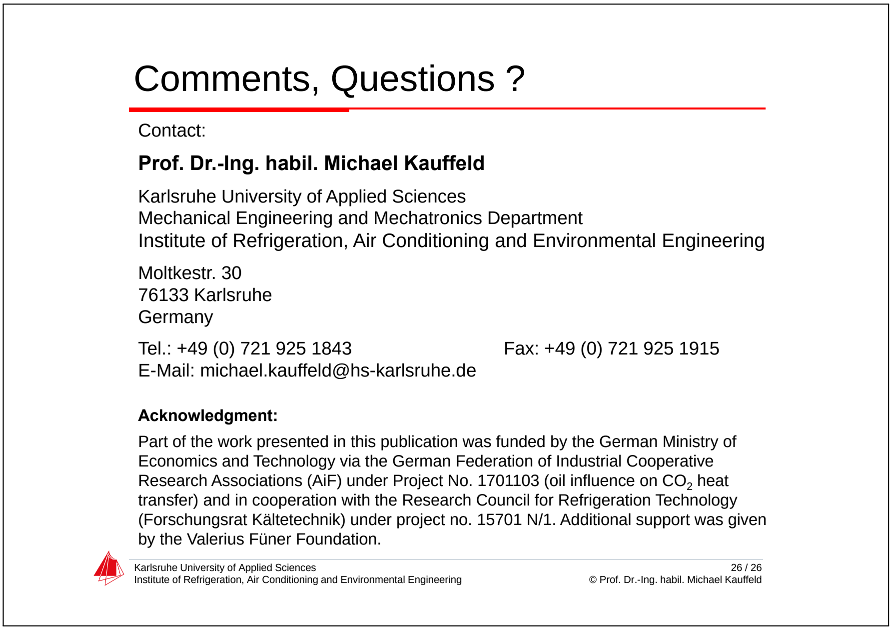
400 W refrigeration capacity with 40 x 40 x 40 mm heat exchanger

Minichannel Heat Exchangers

… offer many advantages

- Higher energy efficiency
- Reduced refrigerant charge and air side pressure drop
- Weight and space reduction
- Easy recycling and lower metal prices
- Improved corrosion resistance
- Several ways for optimization
  - Refrigerant distribution
  - Header tubes
  - Condensate management
  - Profile geometry
  - etc.



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2.2 DTU International Energy Report

Hans Hvidtfeldt Larsen *(hala@dtu.dk)*
DTU National Laboratory for Sustainable Energy

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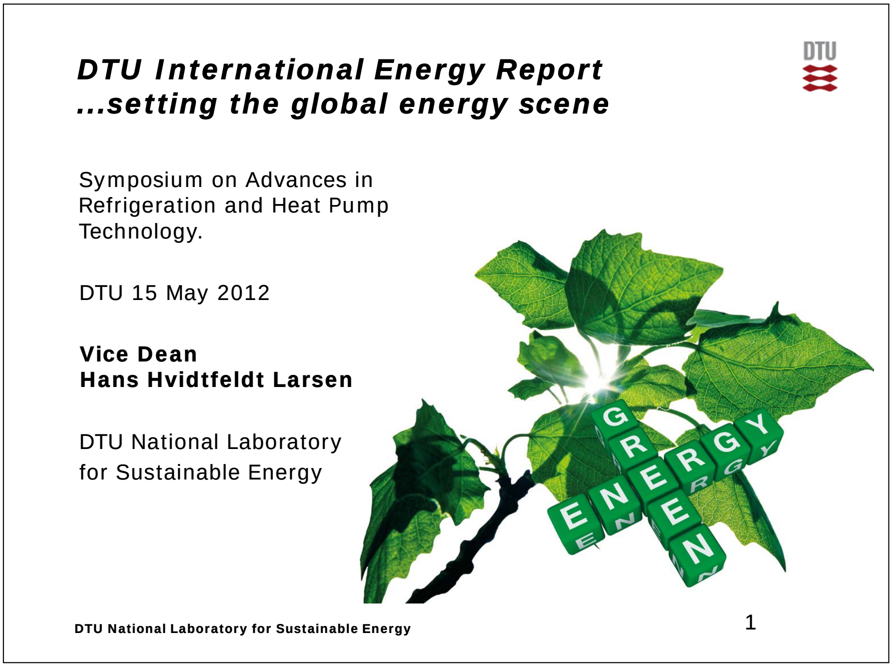

The changing global energy scene

- In the coming years the world is facing major challenges, to ensure the supply of energy for a growing population, particularly in developing countries.
- We need a paradigm shift in the energy systems, we have to move from a system based on fossil fuels to a sustainable non-fossil system.
- Especially in developing countries the challenges are significant and at the same time the possibilities enormous for development and green growth

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The 2^0 target

To keep the global mean temperature rise below 2^{0}C we need to reach global stabilisation at 450 ppm CO_2eq

The economic recession may be seen as short term relief with regard to GHG emissions. Nevertheless:

This means that global greenhouse gas (GHG) emissions must be halved by 2050 and in fact reduced even more in the OECD countries, maybe 80 %

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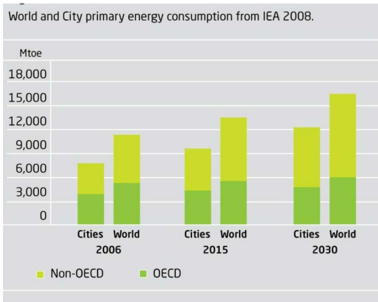
3

Rapid urbanisation

By 2050 more than 6 billion people will live in urban areas, most of them in developing and less-developed countries.

The number of megacities is expected to increase from three in 1975 to 29 by 2025.

At the same time, urbanization generally leads to increased demand for energy-consuming services such as housing and transport.

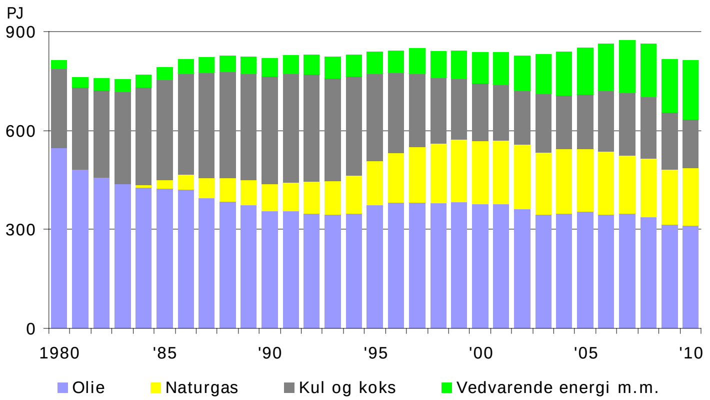


Northern Europe

- Norway is to a very high degree based on hydro power, and has huge oil and gas reserves. For the future the country is aiming at developing its renewable potential like off shore wind.
- Sweden has based its electricity production on a combination of nuclear and hydro power.
- Germany is using nuclear, coal and gas as well as a large amount of wind - more than 18 GW installed in 2010.
- All these countries are interlinked with Denmark and trades continuously electricity.

Danish energy consumption has been stable over the last 30 years

- Is it possible to continue ...?



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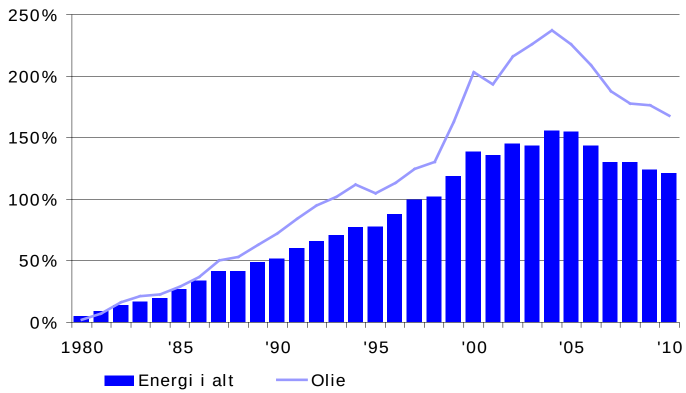
Degree of Self-sufficiency



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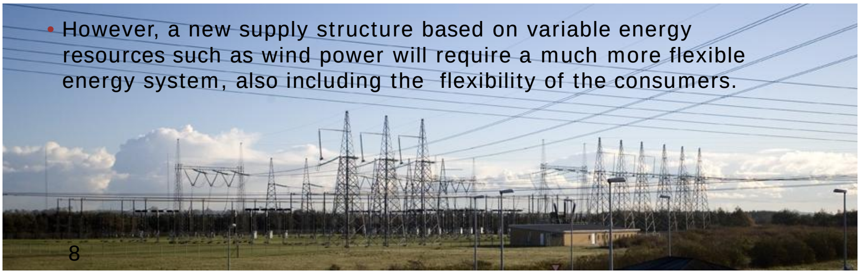
The energy system

- Today's energy system is the result of decisions taken over more than a century.
- This long-term development is reflected in the structure of the energy system, which in most cases was developed according to basic engineering requirements: energy is produced to meet the needs of consumers.
- However, a new supply structure based on variable energy resources such as wind power will require a much more flexible energy system, also including the flexibility of the consumers.



8

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System aspects

It will not be possible to develop the energy systems of the future simply by improving the components of existing systems.

An integrated approach is needed that will optimise the entire system, from energy production, through conversion to an energy carrier, energy transport and distribution, and efficient end-use.

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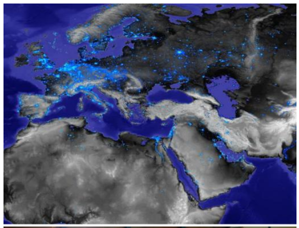
Systems aspects

- Introduce advanced energy storage facilities in the system;
- Integrate the transport sector through the use of plug-in hybrids and electric vehicles;
- Automatic adaptation of consumption to match the availability of energy;
- Development of supergrids interconnecting different regions;
- A smart grid must link production and end-use at the local level;
- Taxes and tariffs should stimulate flexible demand;
- Information and communications technology (ICT) will be very important to the successful integration of renewables in the grid;



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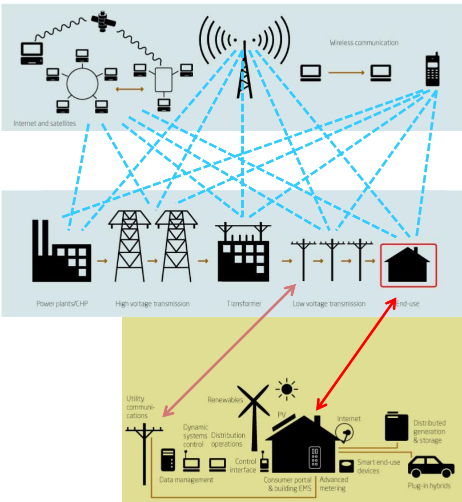
The future intelligent energy system



Information and Communication Technologies

+

+

Distributed generation and efficient building and transport systems

=

The future intelligent energy system emerges

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Future energy supply and end-use technologies

- Further development of sustainable energy supply technologies for use in the intelligent energy system of the future including within smart cities.
- End-use technologies should be developed as active components in the future energy system both for the building sector and the transport sector
- Efficiency improvements should be given high priority in all parts of the energy system

Solar

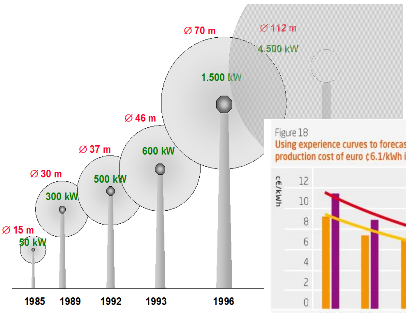
Solar energy can be used for production of heat and electricity all over the world

PV is by nature a distributed generation technology, whereas CSP is a centralised technology

By 2050 PV and CSP technologies will each produce 11% of the world's electricity.

Wind energy

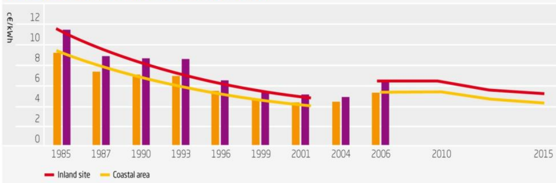
Development of wind turbines



Expected to generate more than 331 TWh in 2010, covering 1.6% of global electricity consumption

Cost of energy from wind and fossil fuels

Figure 18
Using experience curves to forecast wind energy economics up to 2015. The costs shown are for an average 2 MW turbine with a present-day production cost of euro ¢6.1/kWh in a medium wind regime. From [3]



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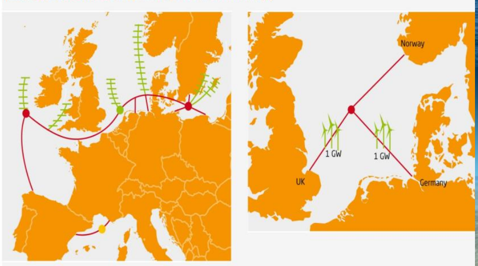
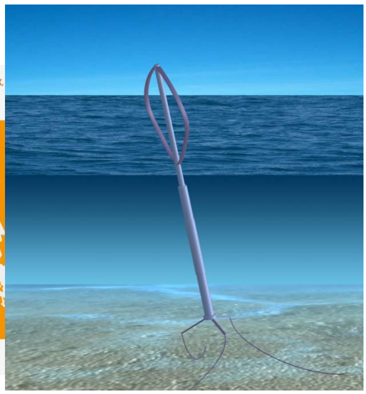
Off-shore wind

Figure 43
The SuperNode configuration could be a first step towards a European supergrid. It would allow the three-way trading of power between the UK, Norway and Germany, and would include two 1 GW offshore wind farms, one in the UK and one in Germany. To balance fluctuations in wind power, up to 1 GW could be transferred between any two of the three countries [6].





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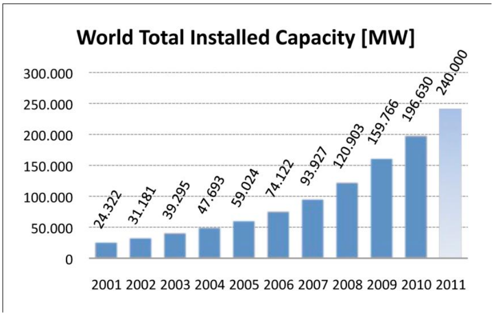

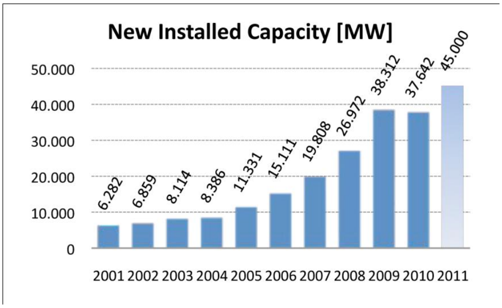

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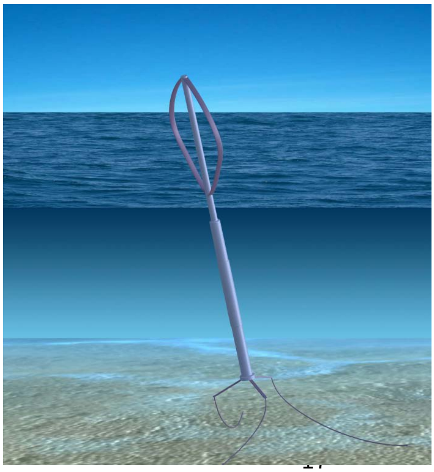
Emerging wind energy technologies

The coming decade may see new technological advances and further scale-up

With increased focus on offshore deployment combined with the radically different conditions compared to onshore, it is likely that completely new concepts will emerge, such as the vertical-axis turbine currently being developed at DTU



Hydro

Hydropower is a mature technology

Wave energy is an interesting partnership with wind energy

Globally, the potential for wave power is at least 10% of total electricity consumption

A goal for Danish wave power by 2050 could be around 5% of electricity consumption

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Biomass

To day 10% of the world's energy consumption. In 2050 up to 200-500 EJ/yr ~ up to half of the world's energy needs in 2050

A large proportion will be wood for direct burning in less developed areas of the world

An easily storable form of energy

Can be used in CHP systems

A source of liquid fuels for transport

A limited resource, and increases in biomass production should not compete with the food supply

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Technology for sustainable energy supply - Bioenergy

- Production and properties of biomass
- Biomass conversion and co-production
- The production of 2 generation bio-fuel from straw by means of an internationally unique method

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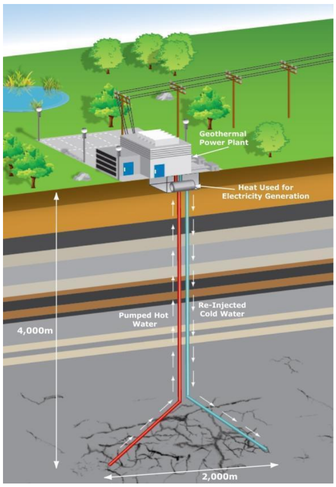
Geothermal

At least 24 countries produce electricity from geothermal energy, while 76 countries use geothermal energy directly for heating and cooling

Global production is 0.2 EJ, with 10 GW of installed baseload electricity production capacity

Potential in 2050 is approximately 200 EJ/yr, of which 65 EJ/yr from electricity production

In Denmark, the potential is substantial and could cover a large part of the demand for future district heating



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Storage

- Energy storage is needed in a future energy system dominated by fluctuating renewable energy.
- Depends on many factors:
  - the mix of energy sources,
  - the ability to shift demand,
  - the links between different energy vectors, and
  - the specific use of the energy.
- Storage costs and energy losses need to be considered.

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CCS

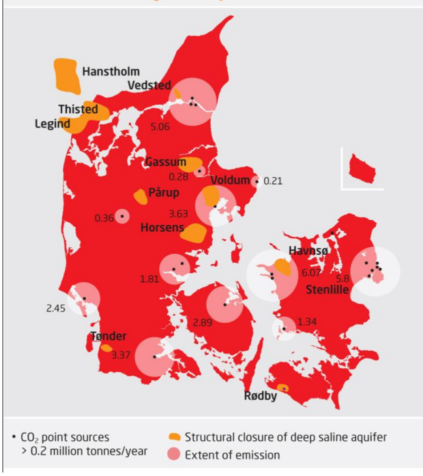
Carbon Capture and Storage (CCS) can be used on large point sources based on fossil fuels such as power plants and industrial furnaces

With CCS we can continue to burn fossil fuels even in a carbon-neutral future

CCS can even be used with biomass-fired power plants to create net negative CO_2 emissions.

Denmark has a good chance of exploiting CCS

Figure 40
The largest potential sites for Danish underground CO_2 storage (excluding hydrocarbon fields in the North Sea), and the largest emitters. Source: Geological Survey of Denmark and Greenland



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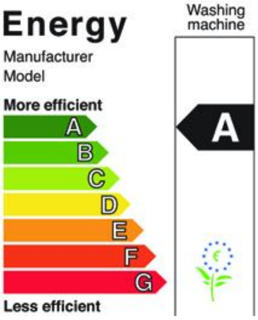
Efficiency improvements

- High emphasis on efficiency improvements in both industry and private households changing demand patterns are going to generate new challenges to system operators and utilities.



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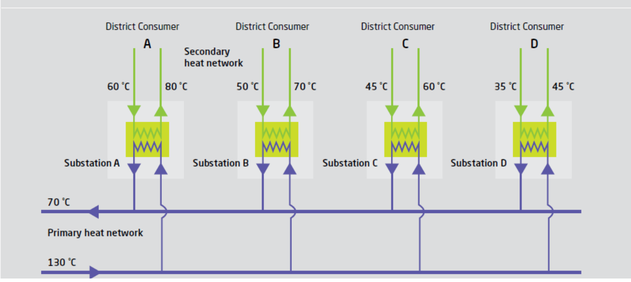
Efficiency improvements through district heating and cooling

- Wider use of district heating would lead to a marked improvement in the efficiency of energy use. Matching up sources of “waste” heat with users of heat, as district heating does, improves overall efficiency and reduces total energy use.
- District heating systems can use surplus power generated by small decentralised renewable resources, such as solar rooftops, in nearby buildings; during periods of high production this energy might otherwise be wasted.

A conventional district heating network.[45]



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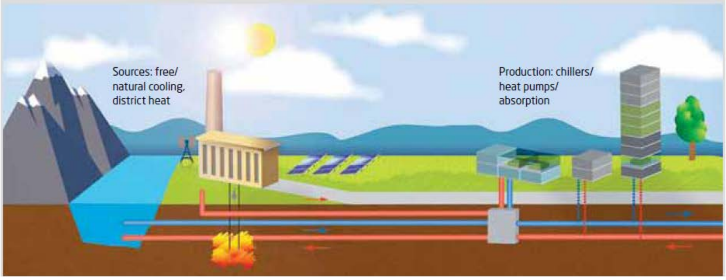
Efficiency improvements through district heating and cooling

- District cooling follows a similar pattern, supplying chilled water and air-condition to a network of users
- Cooling can be obtained directly from seawater or groundwater.
- Alternatively, cooling can be provided via heat pumps or absorption refrigeration systems that take most of their energy from the environment, industrial waste heat or leftover heat from district heating.

Schematic illustration of a district cooling system.



Self sufficient costumers

- The customers are becoming increasingly independent as they in long periods can be self-sufficient with energy by producing some of their limited need for electricity and heat by solar collectors, fuel cells etc.
- In short periods of time they are expecting the system to supply all their needs.

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Transport sector

- Modern transport depends heavily on fossil fuels. Ways to reduce emissions from transport are to shift to renewable or at least CO_2-neutral energy sources, and to link the transpor sector to the power system.

- Achieving this will require new fuels and traction technologies, and new ways to store energy in vehicles.

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Long term development

- Apart from development of the future highly flexible and intelligent energy system infrastructure which facilitates substantial higher amounts of renewable energy than today's energy system
- there is also the need for development of new sustainable supply and end-use technologies for the period after 2050 where CO_2 emissions should be almost eliminated

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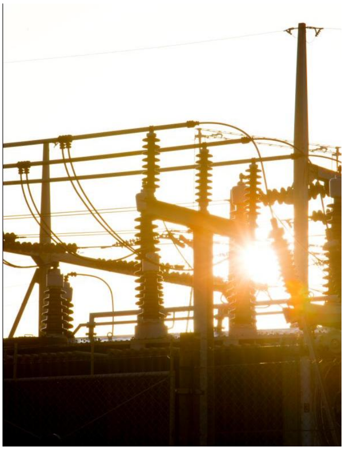
2050

- By 2050, the sum of the potential of all the low-carbon energy sources exceeds the expected demand.
- We need an integrated process that will optimise the entire system, from energy production, through conversion to an energy carrier, energy transport and distribution, and efficient end-use.



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DTU International Energy Report Series

Risø Energy Report 10

Energy for smart cities in an urbanised world

The report is volume 10 in a series that began in 2002

The report addresses energy related issues for smart cities, including energy infrastructure, onsite energy production, transport, economy, sustainability, housing, living and governance, including incentives and barriers influencing smart energy for smart cities.

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Smart cities

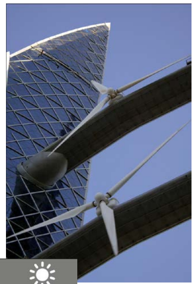
We need a new approach to what cities should do to become more liveable, economically successful, and environmentally responsible:

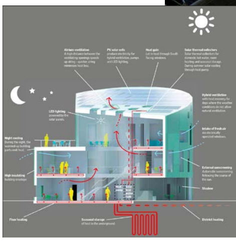
smart cities

that is, energy-efficient, consumer-focused and technology-driven.

Smart buildings

The buildings within a smart city are themselves smart, with internal systems featuring a high degree of interoperability thanks to ICT and connections to the smart grid.





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City energy technologies

A range of renewable energy technologies modified for installation in cities can meet these requirements.

This includes small wind turbines, micro-CHP and heat pumps.

Both solar thermal heating (and cooling) and photovoltaics (PV) are modular technologies that can be integrated in residential, public and commercial buildings.

The production and use of urban biomass should also be promoted.

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Motivation of consumers

A challenge will be to motivate consumers in smart cities to achieve sustainable development by using available technologies in smart ways.

Motivation may be economic, but may also take the form of information, education, regulation, reorganization, or improved services.

Smart technical solutions already exist; now they must be made available to consumers and backed up by suitable economic incentives.

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The creation of a smart city: Make smart buildings

Energy

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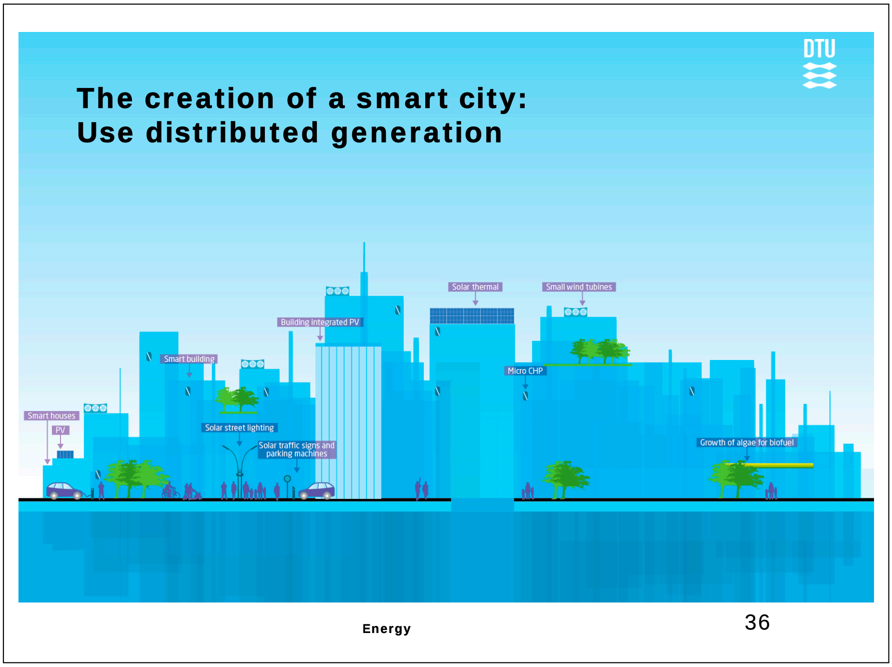
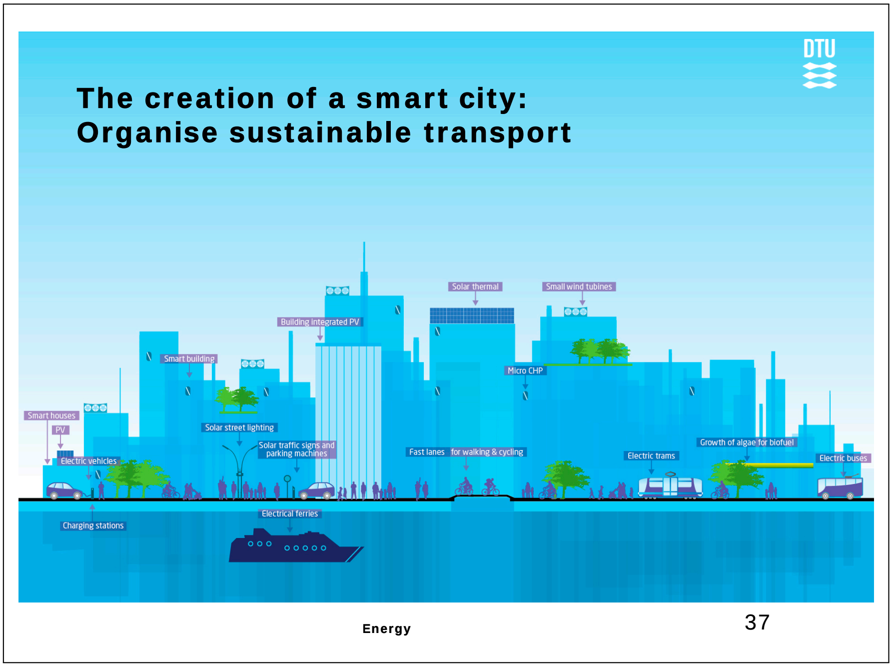
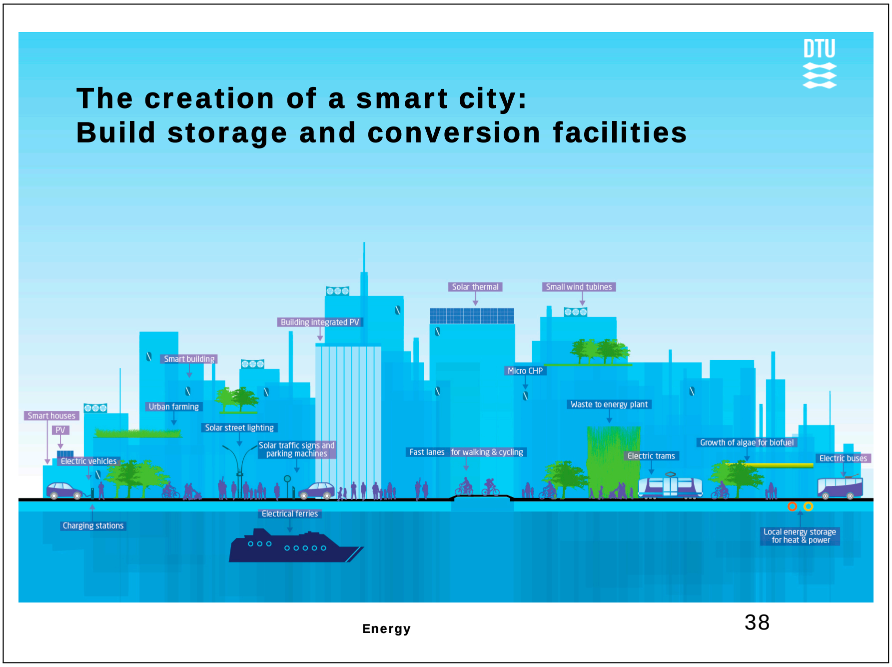
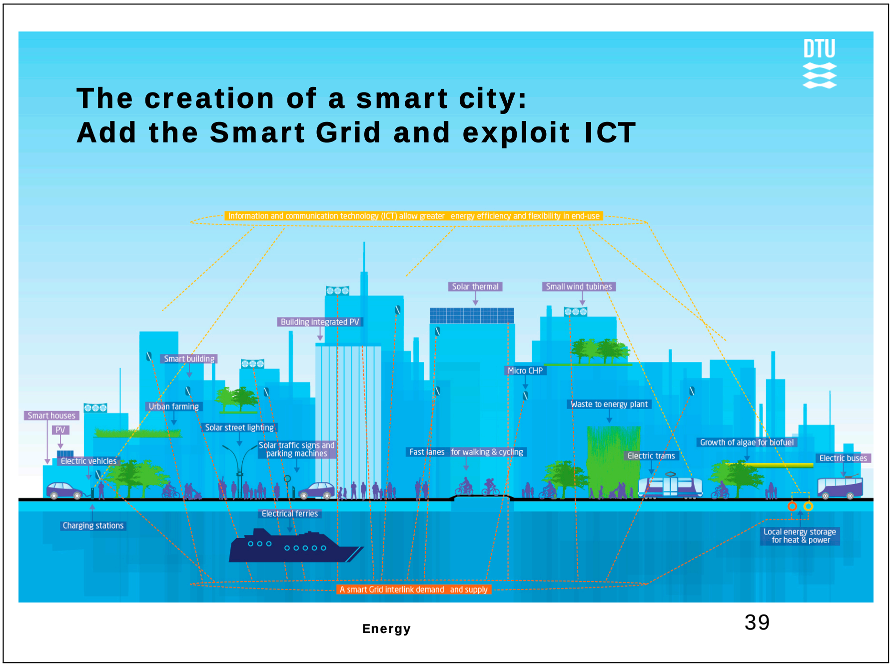

DTU
The creation of a smart city:
Use distributed generation
Solar thermal
Small wind tubines
Building integrated PV
Smart building
Micro CHP
Smart houses
PV
Solar street lighting
Solar traffic signs and parking machines
Growth of algae for biofuel
Energy
36

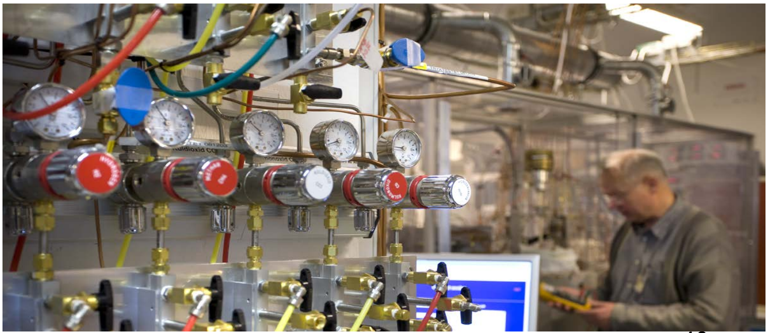

DTU
The creation of a smart city:
Organise sustainable transport
Solar thermal
Small wind tubines
Building integrated PV
Smart building
Micro CHP
Smart houses
PV
Electric vehicles
Solar street lighting
Solar traffic signs and parking machines
Fast lanes for walking & cycling
Electric trams
Growth of algae for biofuel
Electric buses
Charging stations
Electrical ferries
Energy
37


DTU
The creation of a smart city:
Build storage and conversion facilities
Solar thermal
Small wind tubines
Building integrated PV
Smart building
Micro CHP
Urban farming
Smart houses
PV
Waste to energy plant
Solar street lighting
Solar traffic signs and parking machines
Fast lanes for walking & cycling
Electric trams
Growth of algae for biofuel
Electric vehicles
Electric buses
Electrical ferries
Charging stations
Local energy storage for heat & power
Energy
38


DTU
The creation of a smart city:
Add the Smart Grid and exploit ICT
Information and communication technology (ICT) allow greater energy efficiency and flexibility in end-use
Solar thermal
Small wind tubines
Building integrated PV
Smart building
Micro CHP
Urban farming
Smart houses
PV
Waste to energy plant
Solar street lighting
Solar traffic signs and parking machines
Fast lanes for walking & cycling
Electric trams
Growth of algae for biofuel
Electric vehicles
Electric buses
Electrical ferries
Charging stations
Local energy storage for heat & power
A smart Grid interlink demand and supply
Energy
39

Long-term research

- There is a strong need to pursue long-term research and demonstration projects on new energy supply technologies, end-use technologies, and overall systems design. Existing research programmes in these areas should be redefined and coordinated so that they provide the best contribution to the goal of a future intelligent energy system.



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Thank you for your attention

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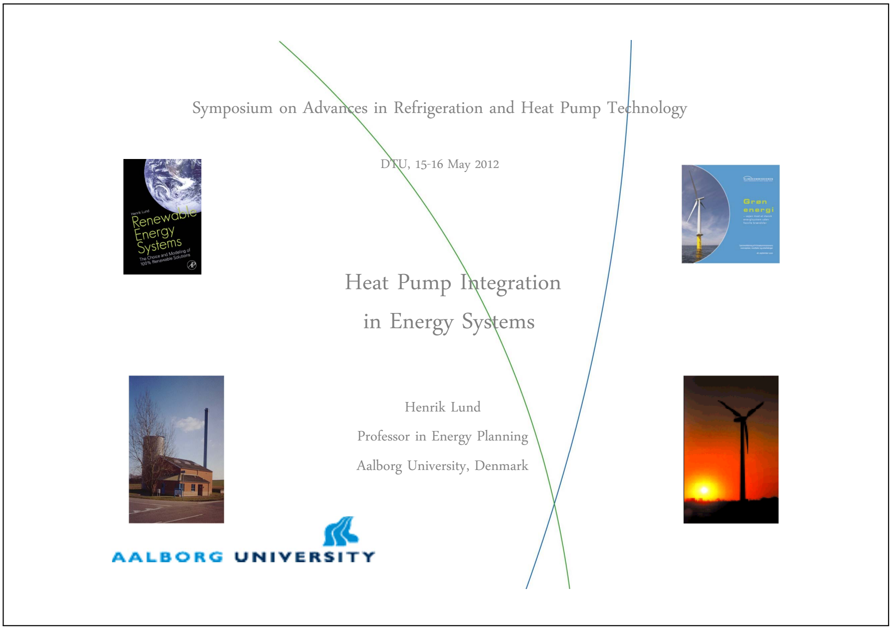
Go back to the table of contents ⏶ or to the timetable ▲

2.3 Heat Pump Integration in Energy Systems

Henrik Lund *(lund@plan.aau.dk)*
Aalborg University

Timetable ▲
Table of contents ⏶



The role of Heat Pumps in the transformation to 100% Renewable Energy Systems

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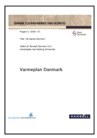
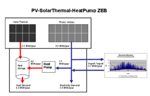
Content

- What kind of Future Energy Systems..?
- How should the buildings be heated..?
  - in order to fit best into the system
  (small versus large-scale heat pumps)
- How should the regulation of Heat Pumps be integrated into the energy system..?





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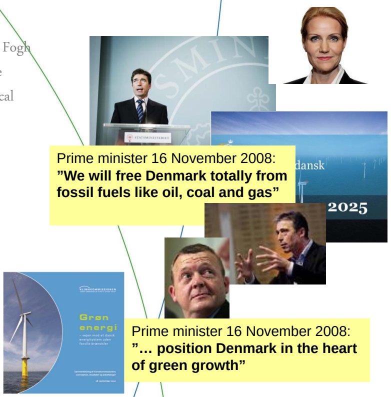
The long-term Objective of Danish Energy Policy

Expressed by former Prime Minister Anders Fogh Rasmussen in his opening speech to the Parliament in 2006 and in several political agreements since then:

To convert to 100% Renewable Energy



Prime minister 16 November 2008:
"We will free Denmark totally from fossil fuels like oil, coal and gas"

Prime minister 16 November 2008:
"… position Denmark in the heart of green growth"

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New Government September 2011

- 100% RES by 2050
- 100% RES for electricity and heating by 2035
- No coal on power plants and no oil for heating households by 2030
- 50% wind in electricity supply by 2020
- 40% CO_2 reduction by 2020 compared to 1990

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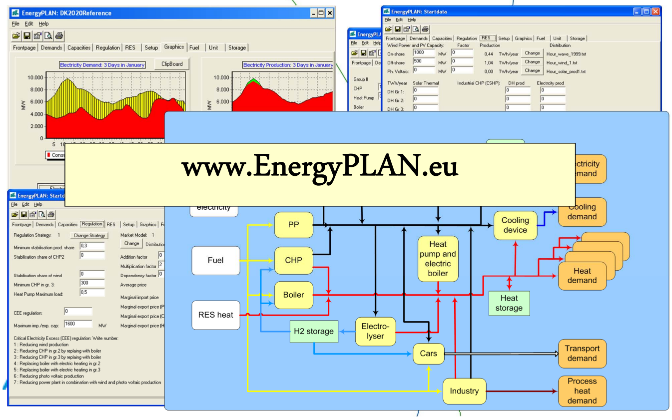
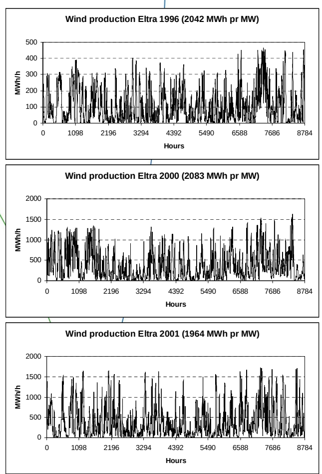
Energi System Analyse Model



Wind energy

Input:

- Data from total productions of wind turbines in the TSO Eltra area (West Denmark).



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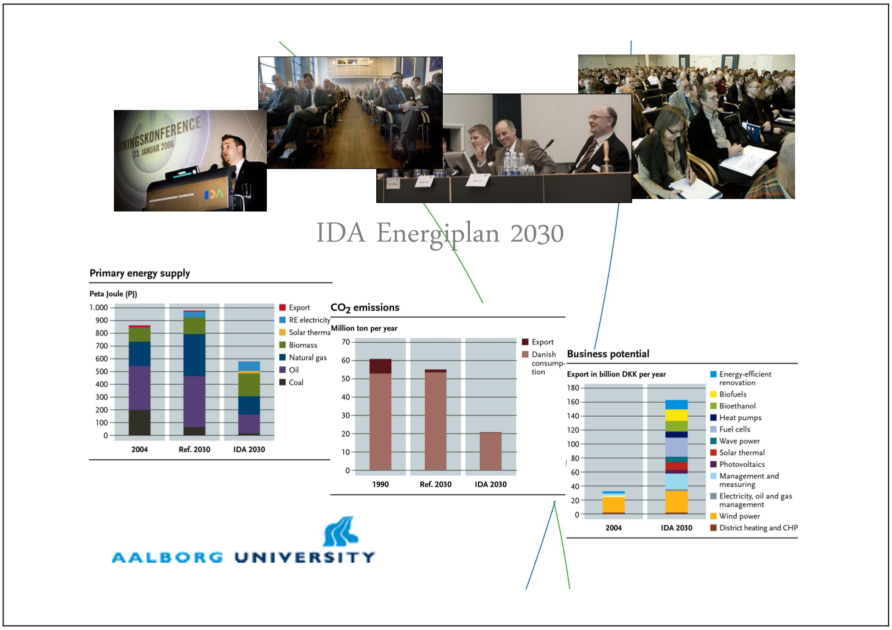
IDA Energiplan 2030
Primary energy supply
Peta Joule (PJ)
1.000
900
800
700
600
500
400
300
200
100
0
2004
Ref. 2030
IDA 2030
Export
RE electricity
Solar thermal
Biomass
Natural gas
Oil
Coal
CO_2 emissions
Million ton per year
70
60
50
40
30
20
10
0
1990
Ref. 2030
IDA 2030
Export
Danish consump-tion
Business potential
Export in billion DKK per year
180
160
140
120
100
80
60
40
20
0
2004
IDA 2030
Energy-efficient renovation
Biofuels
Bioethanol
Heat pumps
Fuel cells
Wave power
Solar thermal
Photovoltaics
Management and measuring
Electricity, oil and gas management
Wind power
District heating and CHP
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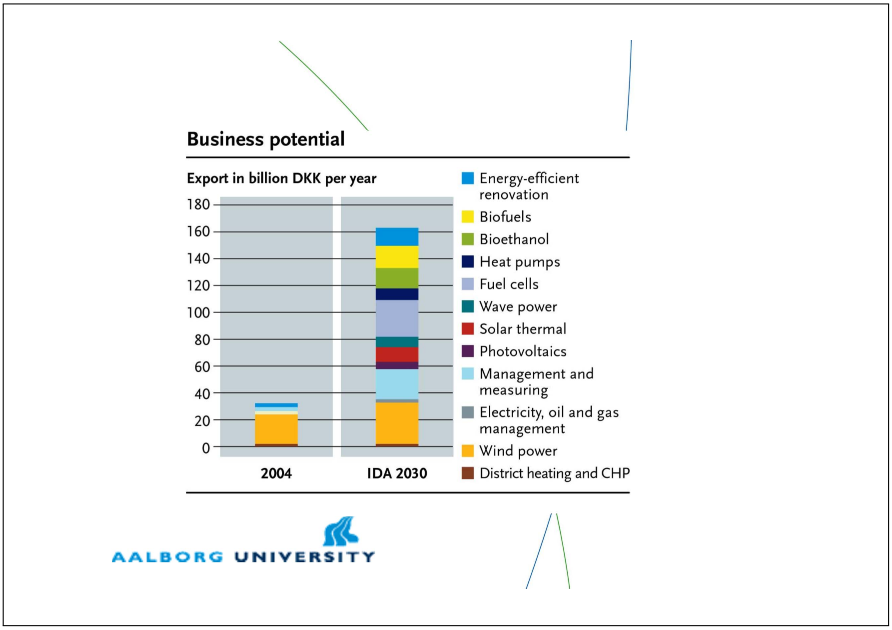
Business potential
Export in billion DKK per year
180
160
140
120
100
80
60
40
20
0
2004
IDA 2030
Energy-efficient renovation
Biofuels
Bioethanol
Heat pumps
Fuel cells
Wave power
Solar thermal
Photovoltaics
Management and measuring
Electricity, oil and gas management
Wind power
District heating and CHP
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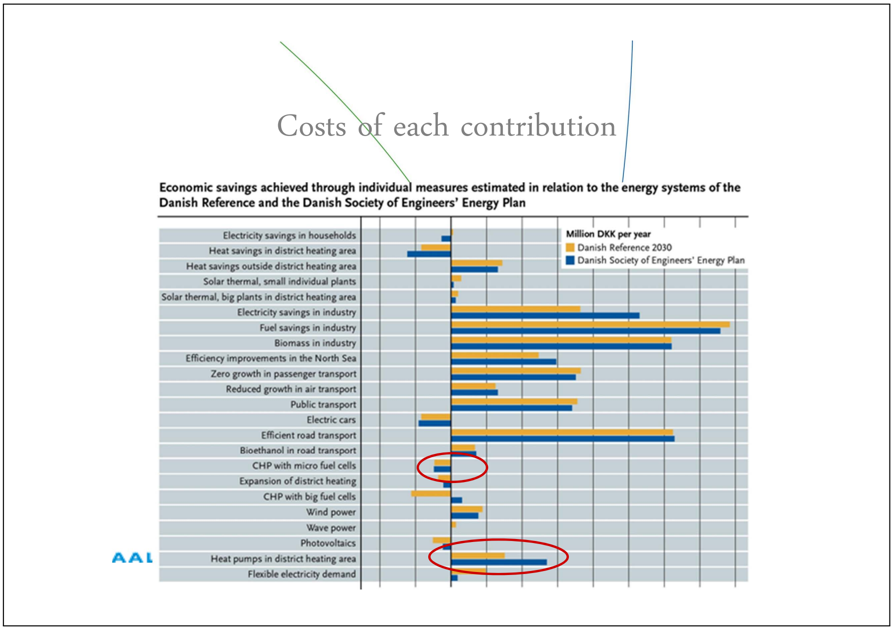
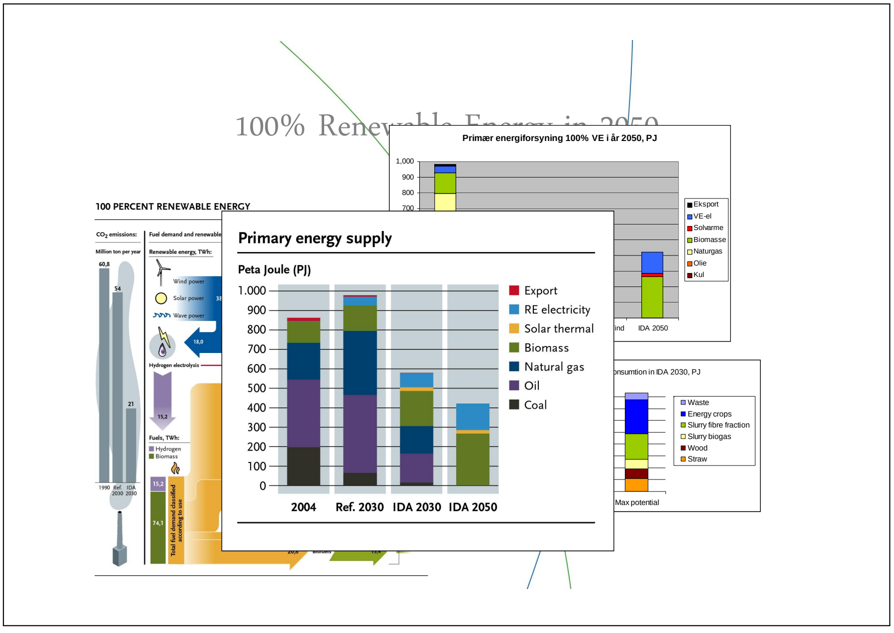

Costs of each contribution
Economic savings achieved through individual measures estimated in relation to the energy systems of the Danish Reference and the Danish Society of Engineers' Energy Plan
Million DKK per year
Danish Reference 2030
Danish Society of Engineers' Energy Plan
Electricity savings in households
Heat savings in district heating area
Heat savings outside district heating area
Solar thermal, small individual plants
Solar thermal, big plants in district heating area
Electricity savings in industry
Fuel savings in industry
Biomass in industry
Efficiency improvements in the North Sea
Zero growth in passenger transport
Reduced growth in air transport
Public transport
Electric cars
Efficient road transport
Bioethanol in road transport
CHP with micro fuel cells
Expansion of district heating
CHP with big fuel cells
Wind power
Wave power
Photovoltaics
Heat pumps in district heating area
Flexible electricity demand
AAL


100% Renewable Energy in 2050
100 PERCENT RENEWABLE ENERGY
Primær energiforsyning 100% VE i år 2050, PJ
Eksport
VE-el
Solvarme
Biomasse
Naturgas
Olie
Kul
IDA 2050
Primary energy supply
Peta Joule (PJ)
1.000
900
800
700
600
500
400
300
200
100
0
2004
Ref. 2030
IDA 2030
IDA 2050
Export
RE electricity
Solar thermal
Biomass
Natural gas
Oil
Coal
Waste
Energy crops
Slurry fibre fraction
Slurry biogas
Wood
Straw
Max potential

Conclusions

- 100 Percent Renewable is **physically possible** and the first toward 2030 is **feasible** to the Danish Society.
- The methodology of design is a very complex process. The combination of a **creative phase** involving many single experts and **detailed system analyses** seems efficient and can be recommended.

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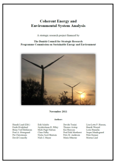
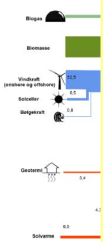
CEESA Project 2011/2012





Transport:
Electric vehicles is best from an energy efficient point of view. But gas and/or liquid fuels is needed to transform to 100%.

Biomass:
.. is a limited resource and can not satisfy all the transportation needs.

Consequence
… Electricity from Wind (and similar resources) needs to be converted to gas and liquied fuels in the long-term perspective…

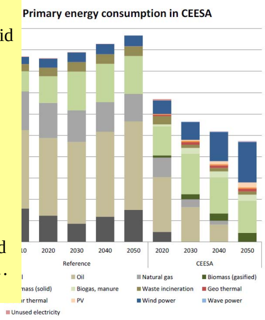
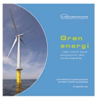


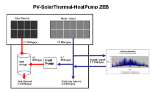
Figure 2: Primary Energy Supply in CEESA.



Content

- What kind of Future Energy Systems..?
- How should the buildings be heated..?
  - in order to fit best into the system
  (small versus large-scale heat pumps)

- How should the regulation of Heat Pumps be integrated into the energy system..?



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Heat Plan Denmark

2008 and 2010

How should we heat the houses in Denmark ??

What to do in a **short-term** perspective in which we want to decrease CO2-emissions and energy consumptions.

And what to do in a **long-term** perspective in which we want to convert to a 100% Renewable Energy System.

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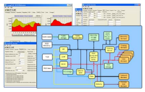
Energy System (Future)



	Nuværende Anno 2006	2020	2040	2060
Tentativ VE-procent				100%
Besparelser på rumvarme	-	25%	50%	75%
Bedre kraftværker	39%	42%	45%	50%
- og kraft/varmeværker	35%/48%	38%/50%	40%/50%	45%/45%
Vindkraft i% af 2006 el-forbrug	16%	33%	50%	75%
El-besparelser i%	-	10%	20%	30%
El-andel af transport	-	10%	30%	50%

Tabel 13-1 En mulig udvikling mo VE system

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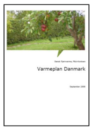
Conclusion



The reasonable solutions seems to be to combine:

- Gradually increasing District Heating from now 46% to somewhere in between 53% and 70%
- Individual Heat Pumps in the rest of the buildings
- Focus on synergies with regards to increasing the efficiencies of district heating networks (essential)

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District Heating because:

- Utilise heat from **waste** used in CHP plants.
- Utilise potentials of **geothermal energy**
- Utilise industrial **excess heat**
- Benefit from **flexible CHP** in combination with heat pumps (better integration of wind power)
- Synergies with regards to **biogas** and **solar thermal**

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District heating helps reducing the pressure on Biomass

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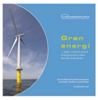
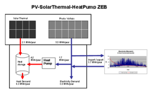


Content

- What kind of Future Energy Systems..?
- How should the buildings be heated..?
  - in order to fit best into the system
    (small versus large-scale heat pumps)

- How should the regulation of Heat Pumps be integrated into the energy system..?



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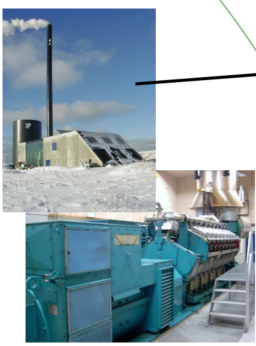
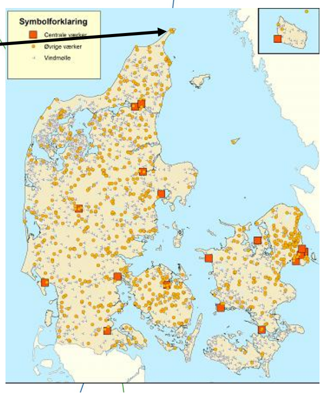
Case: Skagen CHP plant





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Skagen CHP plant

- CHP capacity: 13 MWe and 16 MWth
  (Three 4.3 MWe Wärtsilä Natural Gas engines
- 250 MWh heat storage
- 37 MW peak load boilers
- 10 MW electric boiler
- Heat Pumps Investment under consideration

Operated together with a

Waste Incineration plant (heat only).

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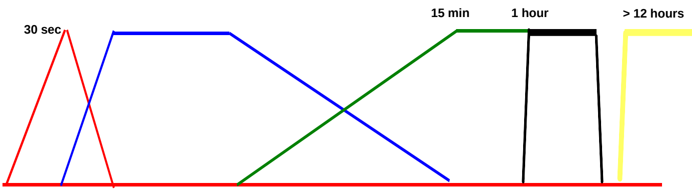
The main electricity markets

- Primary reserves (frequency controlled production)
- Secondary reserves (controlled by status of primary reserves)
- Manuel regulating power (Tertiary reserves)
- Intra day market
- Day ahead spot market



The M.Sc. Programmes in Environm. Managem. & Sustainable Energy Planning and Management, 8. Semester, http://people.plan.aau.dk/~ana

Skagen

- Day ahead spot market in Jan. 2005
- Regulating power market in approx. 2006
- Automatic primary reserve market

  in Nov. 2009

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Cost of entering primary automatic reserves market

- Cost of making +- 1.4 MW available on the engines: Only approx. 27.000 EUR.
- Investing in 10 MW electric boiler:

  Approx 0.7 MEUR.

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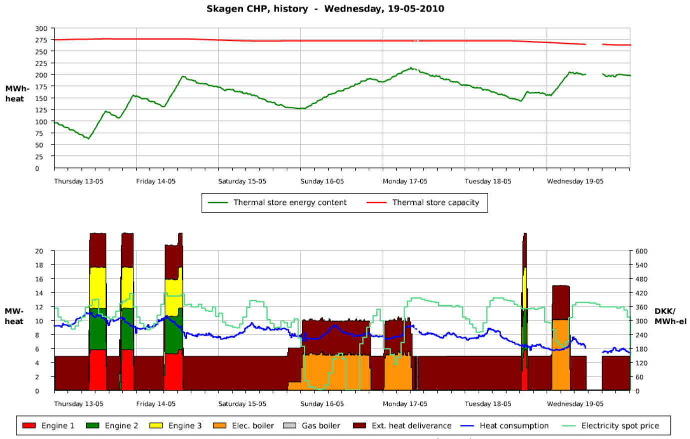

Conclusions

- Denmark can operate a system with 20% Wind and 50% CHP
- By adding heat pumps to the CHP units the integration of wind power can be raised to approx. 40% with-out loosing efficiency (nor wind power)
- Including the CHP plants in the various electricity markets is essential.
- Once the markets are open for CHP plants the cost of entering them seams small.

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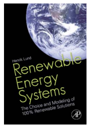
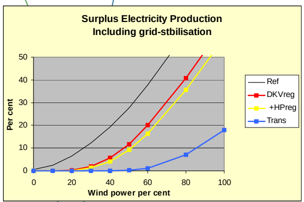


Recommendations I

- Make CHP unit regulation depend on wind power input (10-20% wind without loss of efficiency)
- Add heat pumps (and heat storage capacity) to the CHP units (approx. 40 per cent Wind Power)
- Use electricity for transport as much as possible
- Other kinds of flexible are of less



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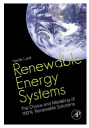
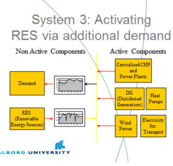
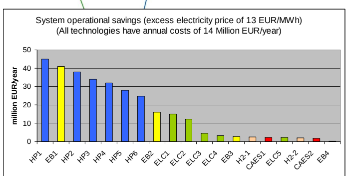


Recommendations II

- Not much gained - (integration of wind nor profit) from investing in electricity storage options
- However the inclusion of CHP, heat pumps and transportation units in securing grid stability is essential.





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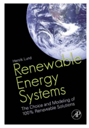
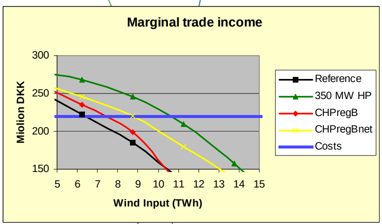


Recommendations III

- The kind of flexibility one need from a technical point of view in a closed system (CHP, HP and transport) is the same kind of flexibility which is needed to raise profits of exchange in an open system.



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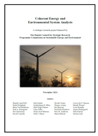
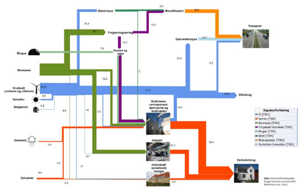


Recommendations IV

- In the medium long term perspective RES electricity has to be transformed into RES gases and liquid fuels (in combination with biomass) to supplement the limited biomass resource. Such conversion opens for the use of gas storage and liquid fuel storage



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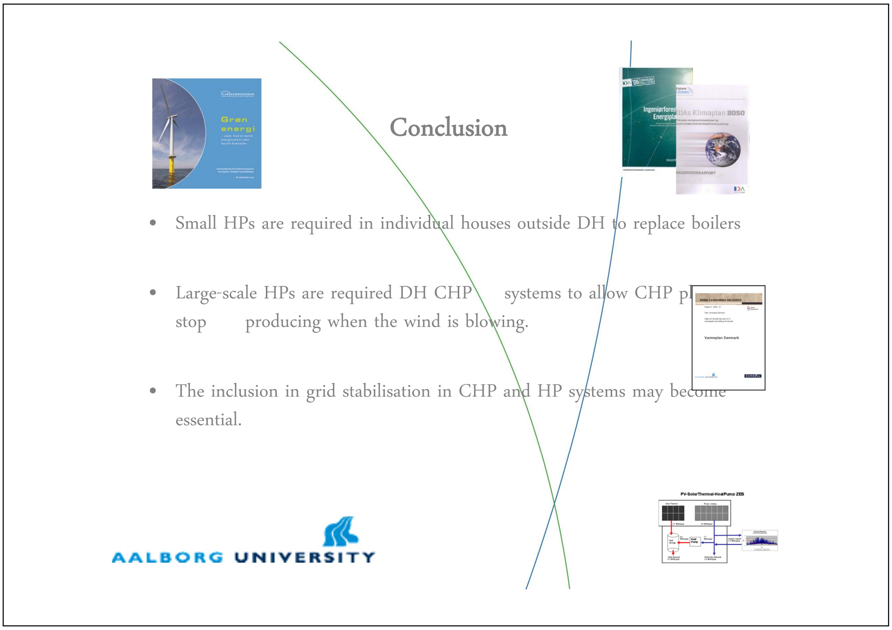
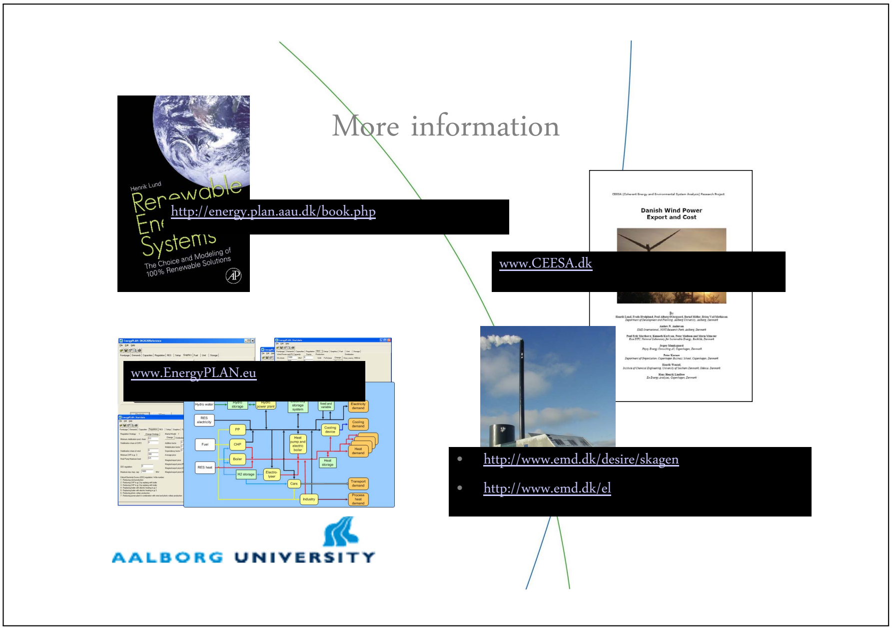


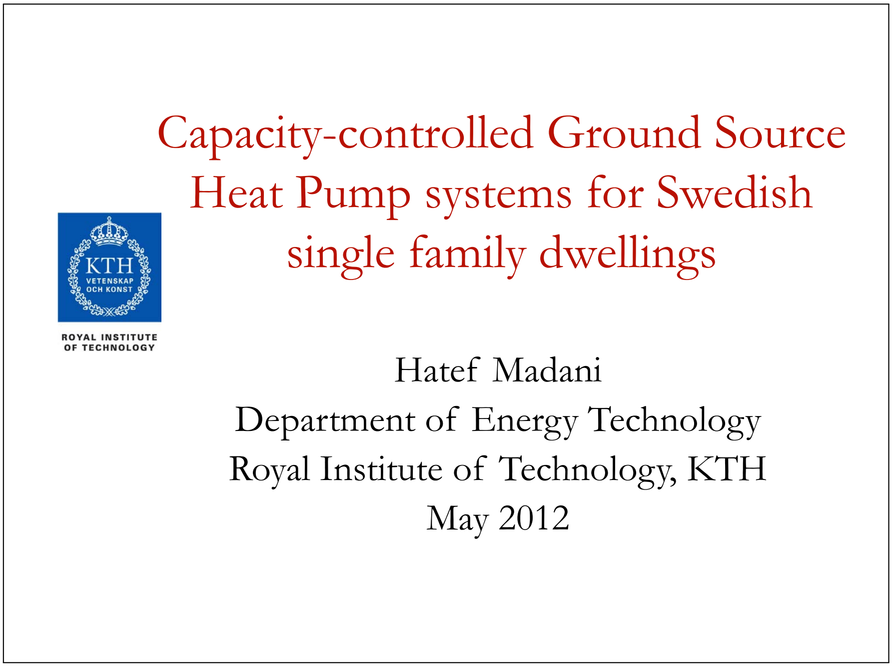


Go back to the table of contents ⯒ or to the timetable ▲

2.4 Capacity-Controlled Ground Source Heat Pumps

Hatef Madani *(hatef.madani@energy.kth.se)*
Royal Insitute Of Technology, KTH, Sweden

Timetable ▲
Table of contents ⊼



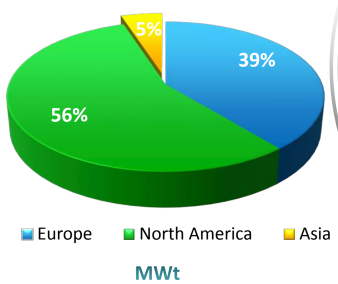
Presentation Agenda

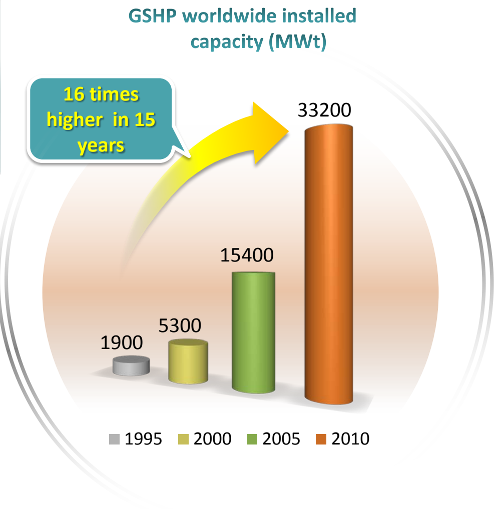
1. Introduction
2. System Approach
3. SWOT Analysis
4. Final remarks

GSHP worldwide capacity-I

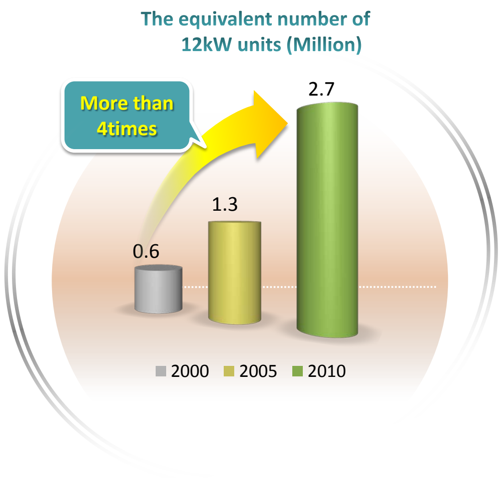
A fast growing market

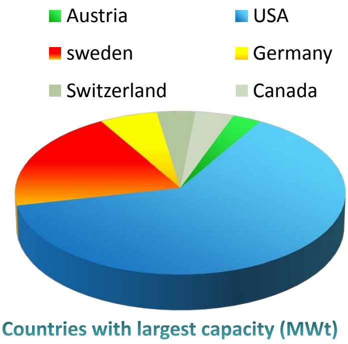
✓33,200 MWt installed capacity



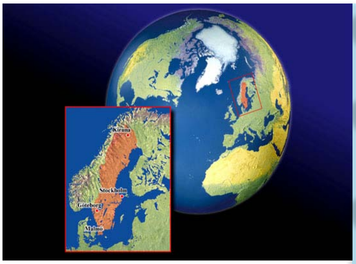


GSHP worldwide capacity-II



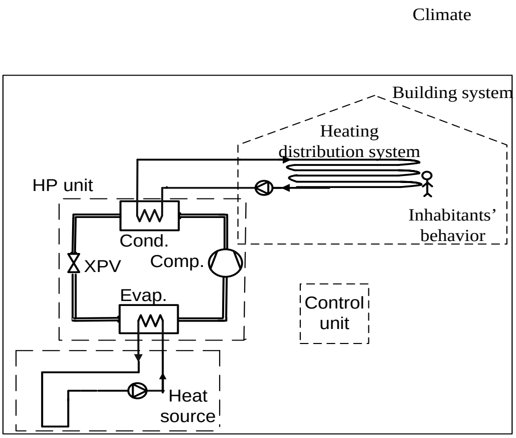


Heat Pumps in single family dwellings: Swedish perspective



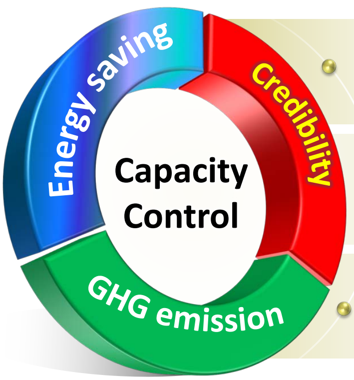
- Share of HP in single family dwellings > 50%
- Share of HP in newly-built single family house > 90%
- Total market value (excluding maintenance and service) is 922 000 000 €
- Approximately 400 000 Brine/Water HP installation
- During 2009, about 3.6 TWh electricity used in order to deliver 11.7 TWh heat to buildings

Why Capacity control ?



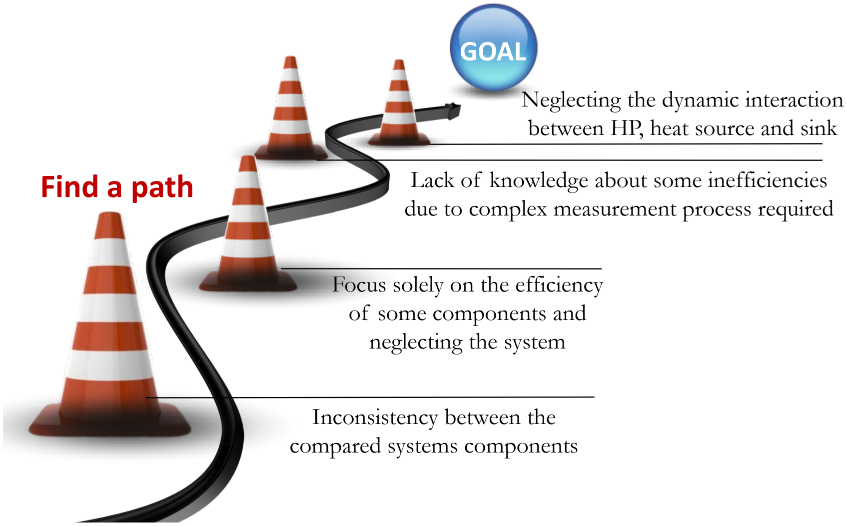
Overall objective

Development of a structured method in order to approach the challenge of better understanding the techniques and potential for efficiency improvement via capacity control in brine to water heat pumps



- Reducing the annual energy use in single family dwellings
- Gain a higher credibility for GSHPs in the market,
- Reduce the greenhouse gas emissions

Barriers and Limitations



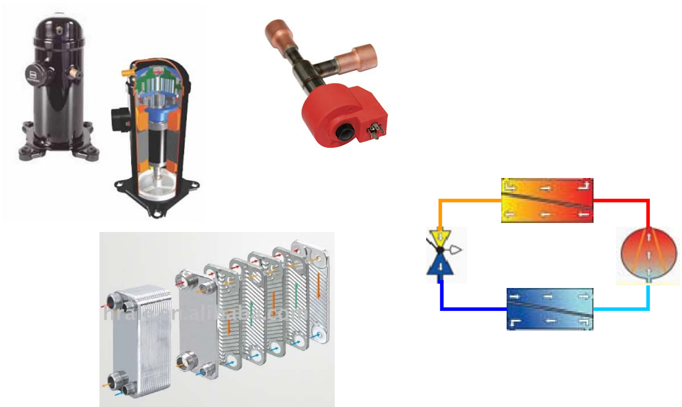
Presentation Agenda

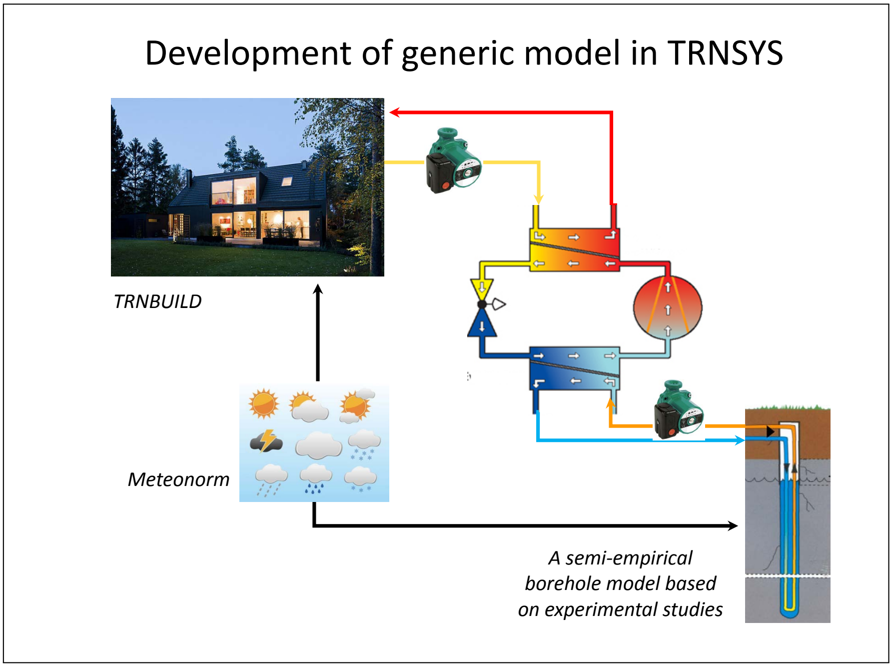
1. Introduction
2. Methodology:

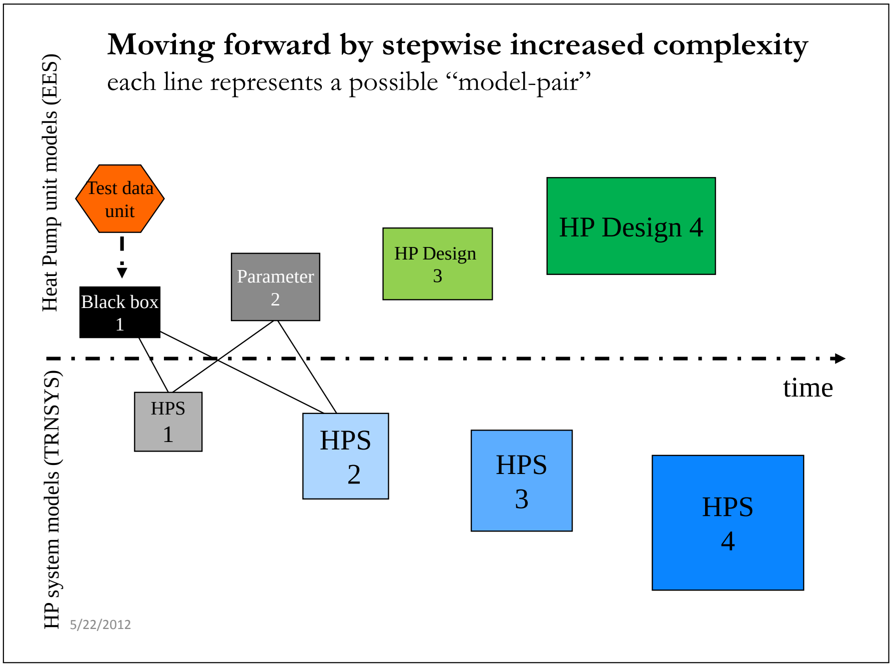
Methodology

1. System approach: Develop a generic model in the presence of heat source, heat sink, etc.
2. Develop the qualitative and quantitative model of the system
3. Validate the models against experimental results
4. Use the generic model to address the questions

Heat pump unit modelling in EES




Development of generic model in TRNSYS
TRNBUILD
Meteonorm
A semi-empirical borehole model based on experimental studies


Moving forward by stepwise increased complexity
each line represents a possible "model-pair"
Heat Pump unit models (EES)
HP system models (TRNSYS)
Test data unit
Black box 1
Parameter 2
HP Design 3
HP Design 4
time
HPS 1
HPS 2
HPS 3
HPS 4
5/22/2012

Presentation Agenda

1. Introduction
2. Methodology
3. Results examples

Results

Some examples are presented in several scientific papers to show how the generic model can facilitate analysis of the annual performance of GSHP systems

Comparative analysis I

between the annual performance of on/off controlled and variable capacity GSHP systems

Comparative analysis II

between three on/off control methods commonly used in European GSHP systems

Descriptive analysis

of GSHP system equipped with variable speed compressor and variable speed pump in U-pipe borehole heat exchanger

Predictive analysis

and evaluation of annual performance of a run-around coil heat recovery system equipped with a variable capacity heat pump unit

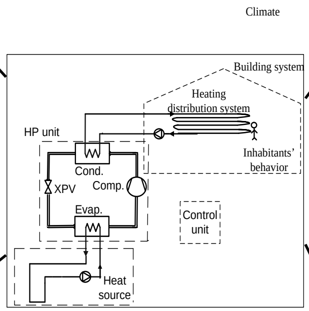
Presentation Agenda

1. Introduction
2. Methodology
3. SWOT Analysis

Variable capacity heat pump system equipped with variable speed compressor and pump SWOT analysis

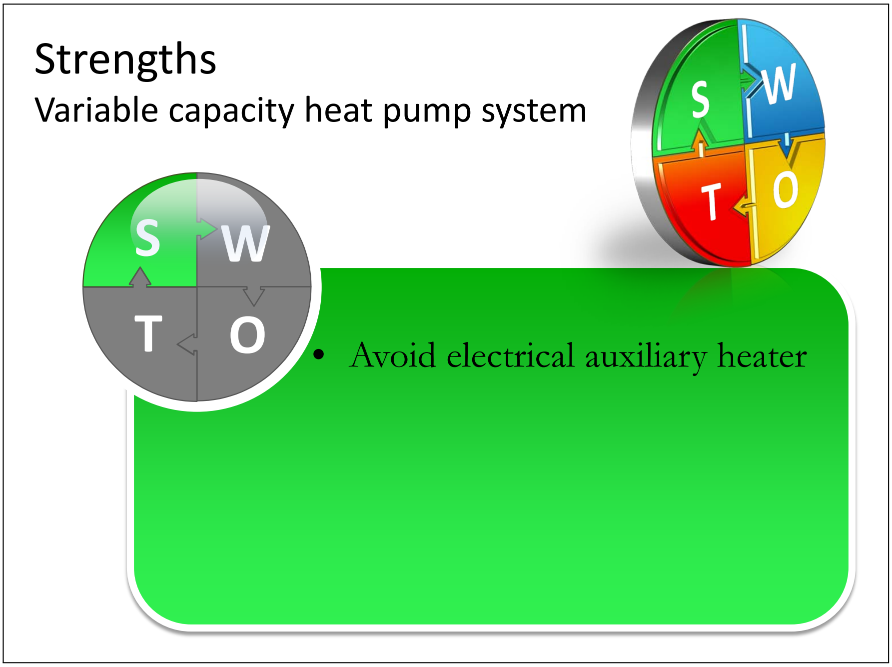
Strengths

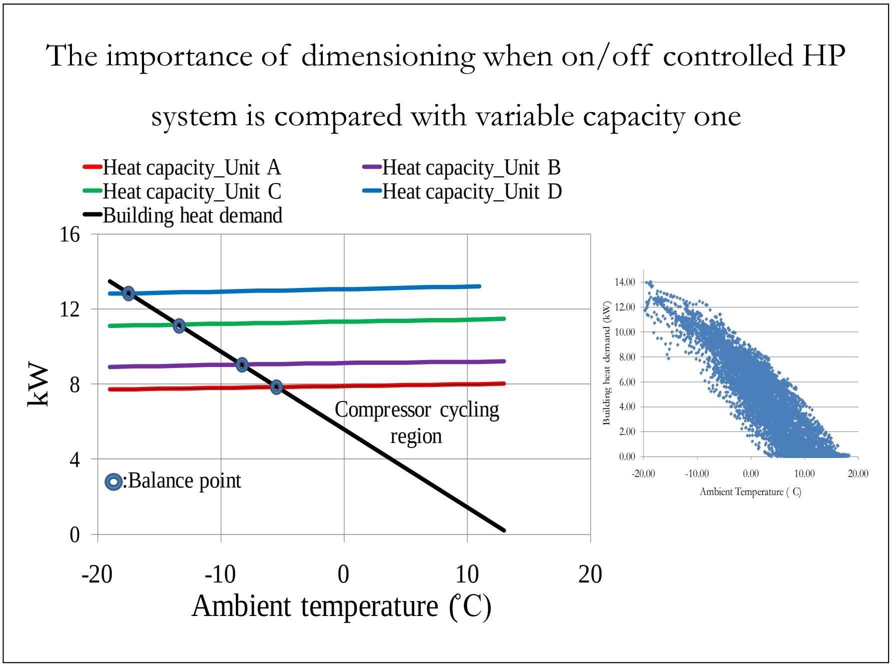
Weaknesses



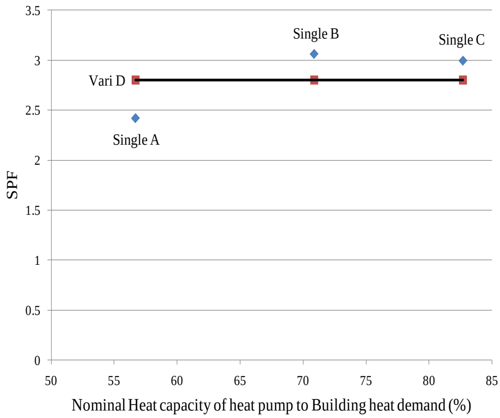
Threats

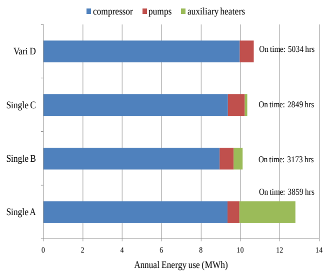
Opportunities


Strengths
Variable capacity heat pump system
S
W
T
O
• Avoid electrical auxiliary heater


The importance of dimensioning when on/off controlled HP system is compared with variable capacity one
Heat capacity_Unit A
Heat capacity_Unit B
Heat capacity_Unit C
Heat capacity_Unit D
Building heat demand
16
12
8
4
0
kW
-20
-10
0
10
20
Ambient temperature (℃)
Compressor cycling region
:Balance point
14.00
12.00
10.00
8.00
6.00
4.00
2.00
0.00
Building heat demand (kW)
-20.00
-10.00
0.00
10.00
20.00
Ambient Temperature (C)

The importance of dimensioning when on/off controlled HP system is compared with variable capacity one

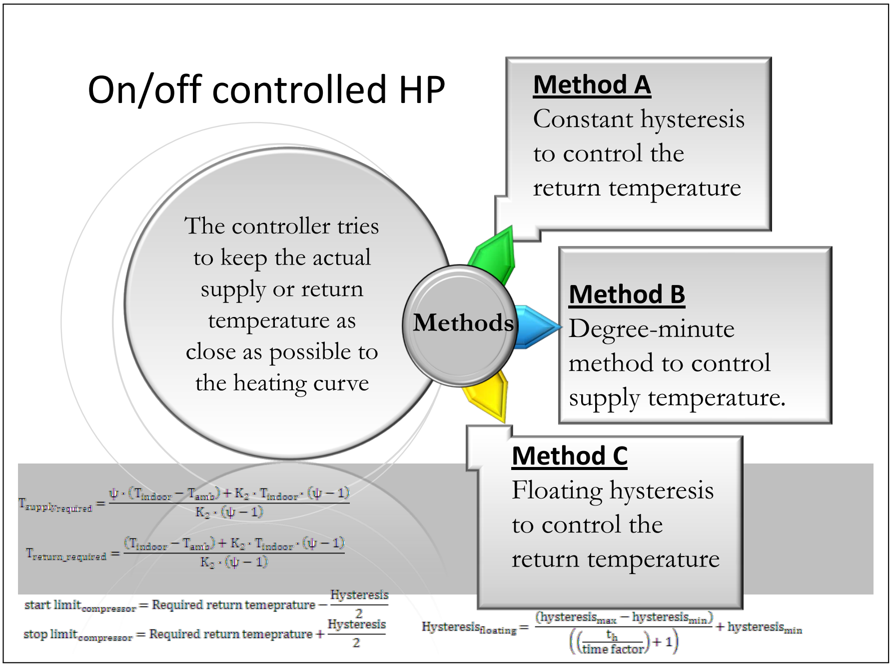


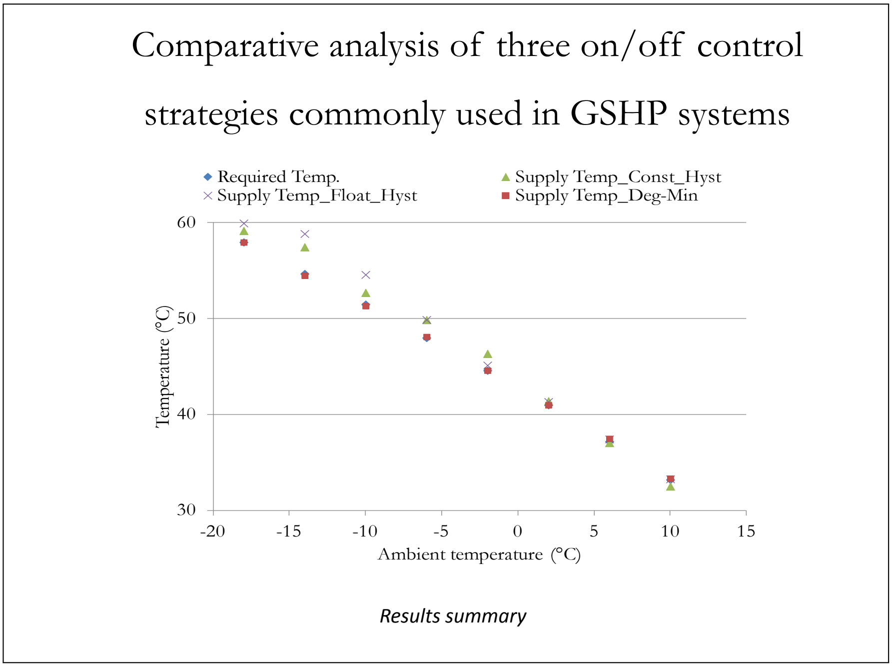


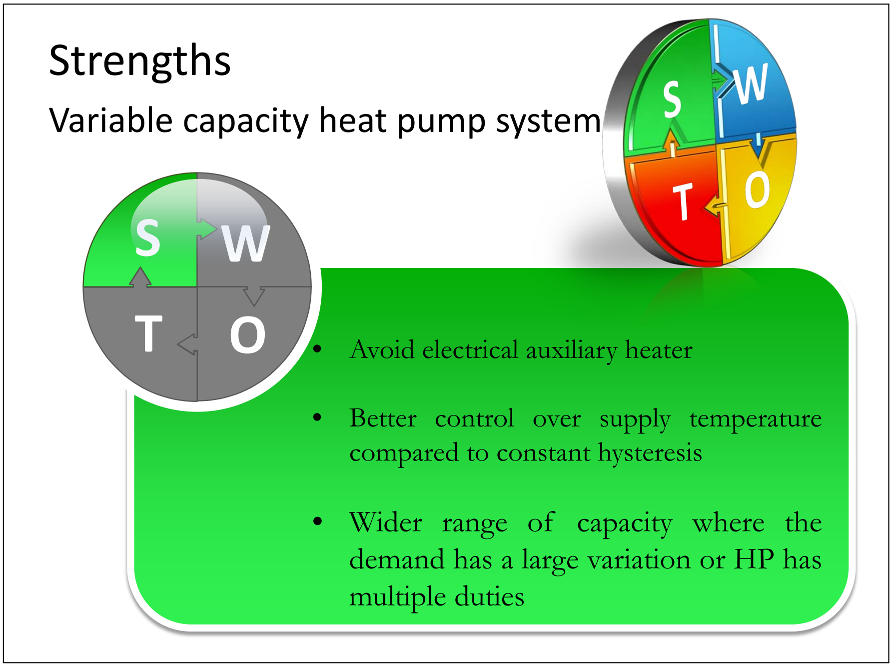
Strengths

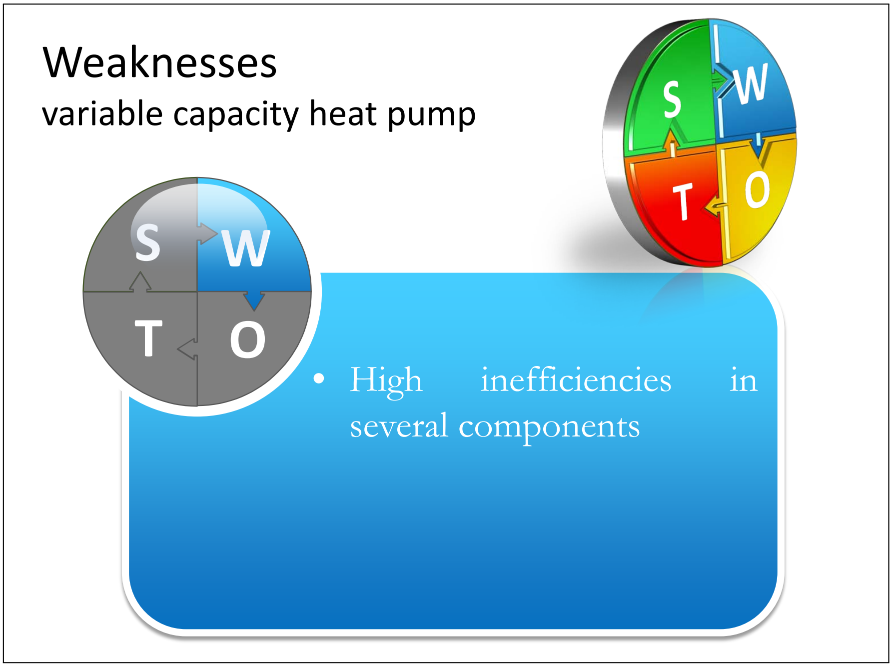
Variable capacity heat pump system

- Avoid electrical auxiliary heater
- Better control over supply temperature?

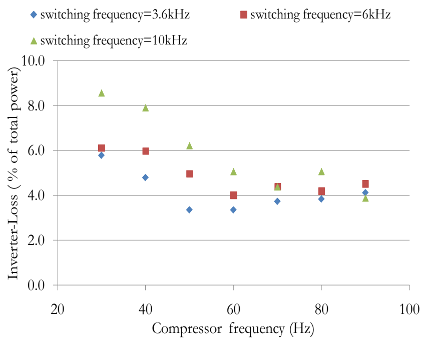

On/off controlled HP
The controller tries to keep the actual supply or return temperature as close as possible to the heating curve
Methods
Method A
Constant hysteresis to control the return temperature
Method B
Degree-minute method to control supply temperature.
Method C
Floating hysteresis to control the return temperature
$T_{supply_{required}} = \frac{\psi \cdot (T_{indoor} - T_{amb}) + K_2 \cdot T_{indoor} \cdot (\psi - 1)}{K_2 \cdot (\psi - 1)}$
$T_{return\_required} = \frac{(T_{indoor} - T_{amb}) + K_2 \cdot T_{indoor} \cdot (\psi - 1)}{K_2 \cdot (\psi - 1)}$
$\text{start limit}_{compressor} = \text{Required return temeprature} - \frac{\text{Hysteresis}}{2}$
$\text{stop limit}_{compressor} = \text{Required return temeprature} + \frac{\text{Hysteresis}}{2}$
$\text{Hysteresis}_{floating} = \frac{(\text{hysteresis}_{max} - \text{hysteresis}_{min})}{\left(\left(\frac{t_h}{\text{time factor}}\right) + 1\right)} + \text{hysteresis}_{min}$


Comparative analysis of three on/off control strategies commonly used in GSHP systems
Required Temp.
Supply Temp_Float_Hyst
Supply Temp_Const_Hyst
Supply Temp_Deg-Min
Temperature (°C)
60
50
40
30
-20
-15
-10
-5
0
5
10
15
Ambient temperature (°C)
Results summary

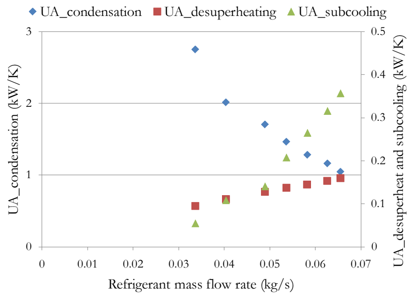

Strengths
Variable capacity heat pump system
S W T O
• Avoid electrical auxiliary heater
• Better control over supply temperature compared to constant hysteresis
• Wider range of capacity where the demand has a large variation or HP has multiple duties


Weaknesses
variable capacity heat pump
S W T O
• High inefficiencies in several components

Inefficiencies in the components when the compressor speed changes in a variable capacity GSHP



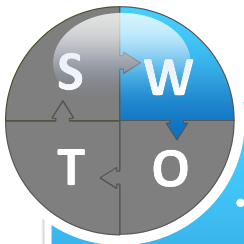
Inverter losses as the percentage of total power



Overall heat transfer coefficient in condenser

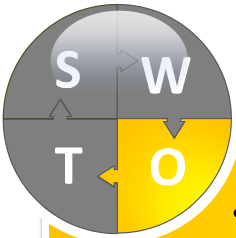
Weaknesses

variable capacity heat pump



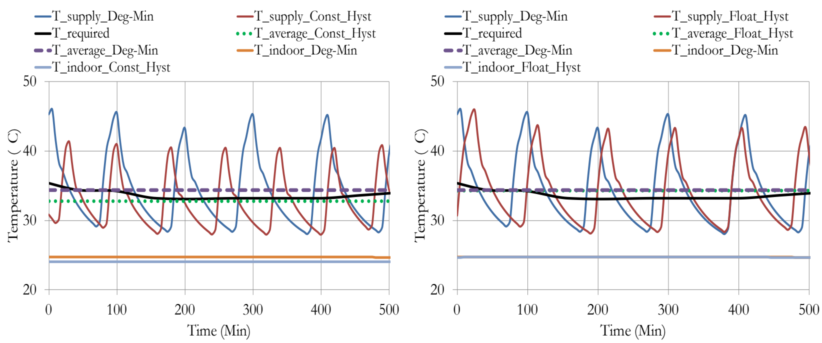
- High inefficiencies in several components
- Cost limitation (almost removed recently)
- Increased complexity

Opportunities



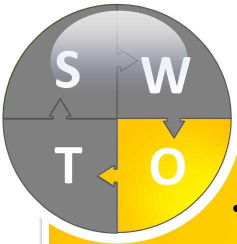
- Optimized control of some parameters in ”conventional on/off control”

Comparative analysis of three on/off control strategies commonly used in GSHP systems (I)



Comparison between the supply temperature and indoor temperature when the ambient temperature changes between +8°C and +10°C.

Opportunities

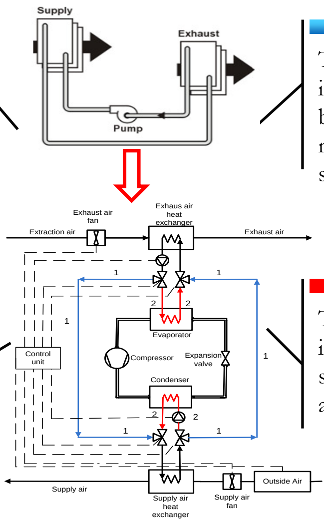


- Optimized control of some parameters in ”conventioal on/off control”
- Innovative way to use frequency inverter
- Finding new market for Heat pump such as heat recovery

Retrofitting a capacity-controlled heat pump to a run-around coil ventilation heat recovery system

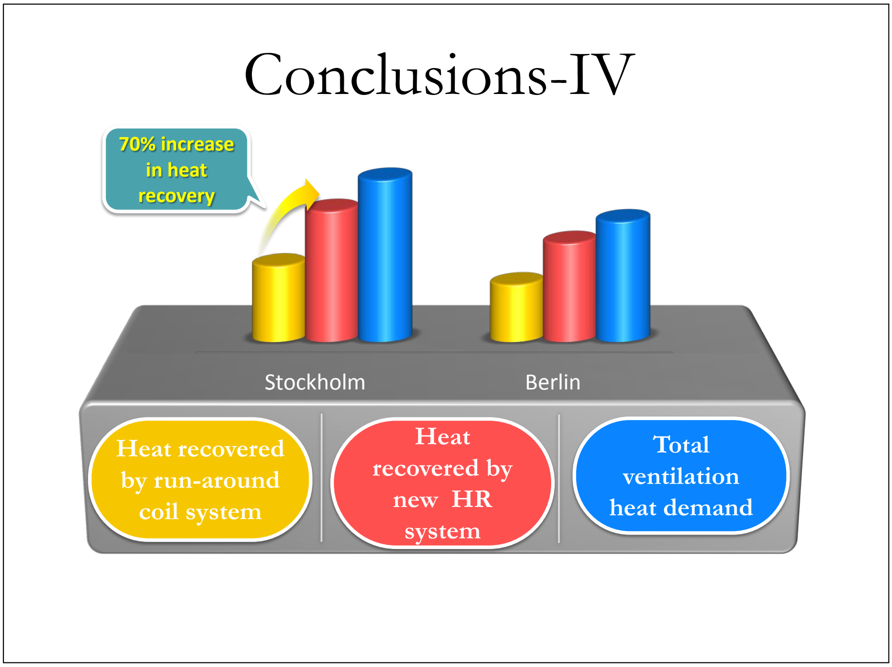
proposing a method for ventilation heat recovery system

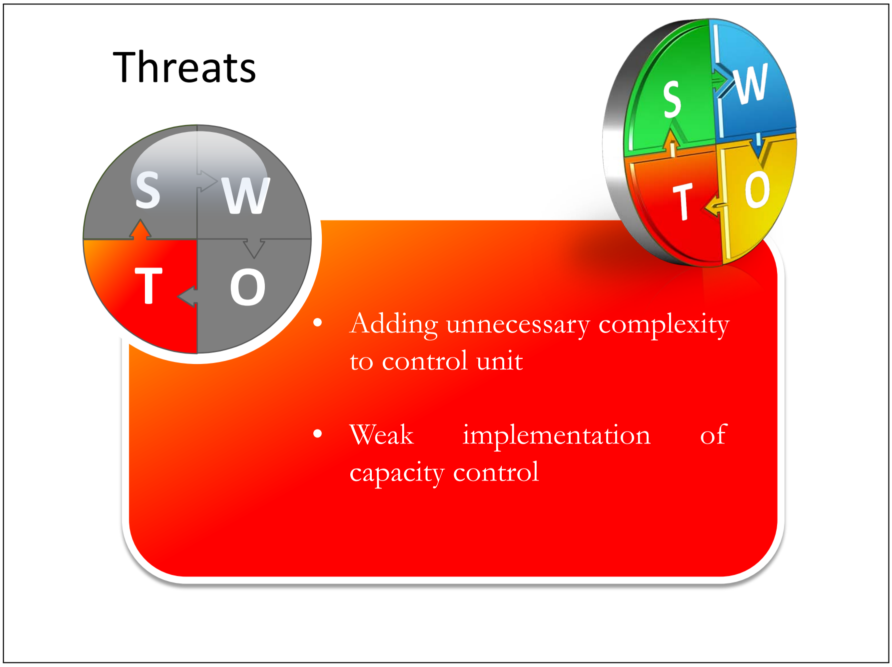
The annual modeling is carried out for both traditional and new heat recovery systems.

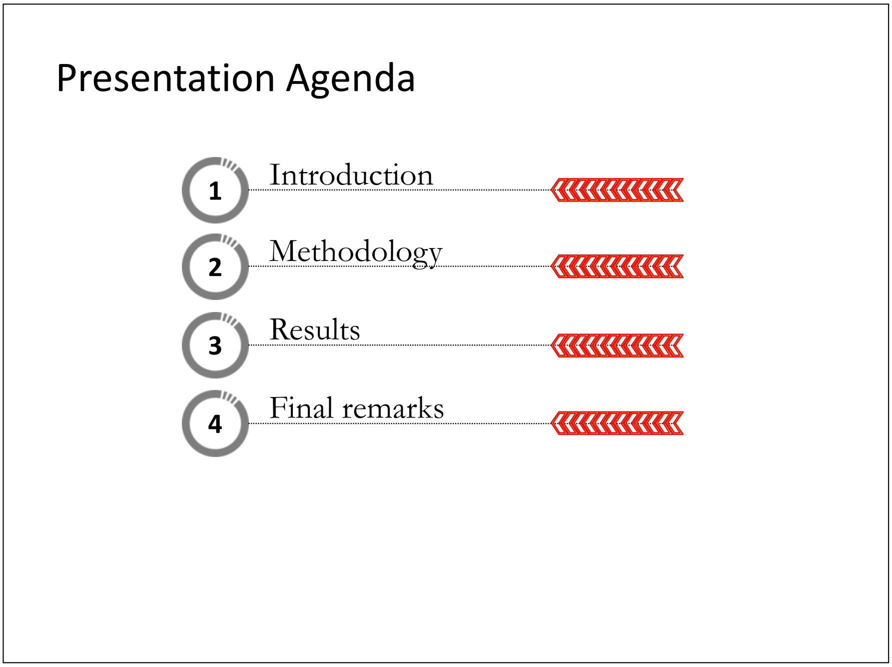


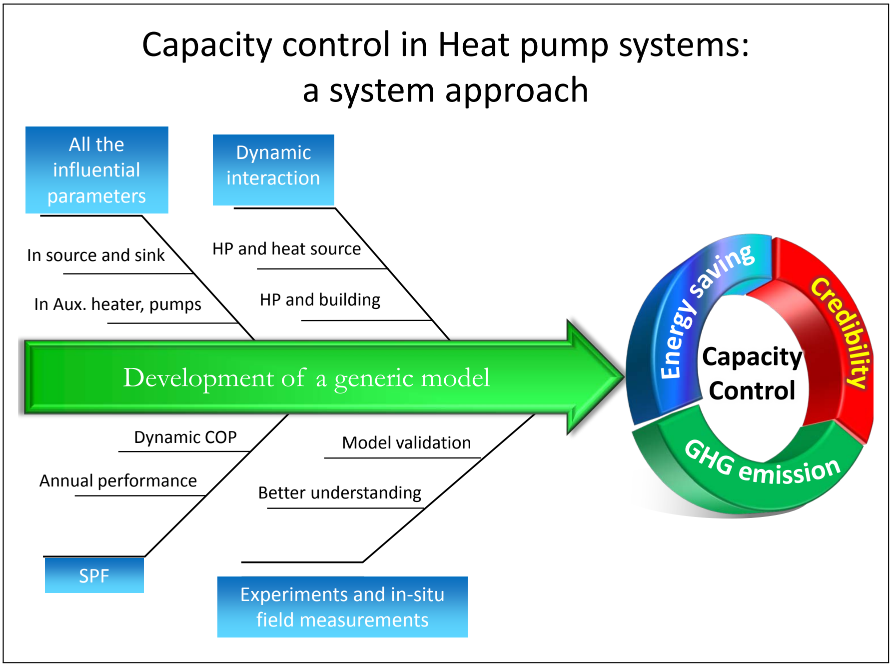
The annual performance of the new systems is compared with the traditional run-around coil system.

The estimated improvement of the system efficiency is analyzed


Conclusions-IV
70% increase in heat recovery
Stockholm
Berlin
Heat recovered by run-around coil system
Heat recovered by new HR system
Total ventilation heat demand


Threats
S
W
T
O
• Adding unnecessary complexity to control unit
• Weak implementation of capacity control


Presentation Agenda
1 Introduction
2 Methodology
3 Results
4 Final remarks


Capacity control in Heat pump systems:
a system approach
All the influential parameters
Dynamic interaction
In source and sink
In Aux. heater, pumps
HP and heat source
HP and building
Development of a generic model
Dynamic COP
Annual performance
SPF
Model validation
Better understanding
Experiments and in-situ field measurements
Energy saving
Credibility
GHG emission
Capacity Control

Publications' examples

- Madani H., Claesson J., Lundqvist P. 2011 "Capacity control in ground source heat pump systems, Part I: modeling and simulation", International Journal of Refrigeration, Volume 34 (6), Issue 6, pp 1338-1347.

- Madani H., Claesson J., Lundqvist P. 2011 "Capacity control in ground source heat pump systems, Part II: Comparative analysis between on/off controlled and variable capacity systems", International Journal of Refrigeration, Volume 34 (8), pp 1934-1942.

- Madani H., Lundqvist P. 2011 "Evaluation of the annual performance of Ground Source Heat Pump systems: A comparison between single speed and variable speed systems", 23rd IIR International Congress of Refrigeration, Prague, Czech Republic, ID 843.

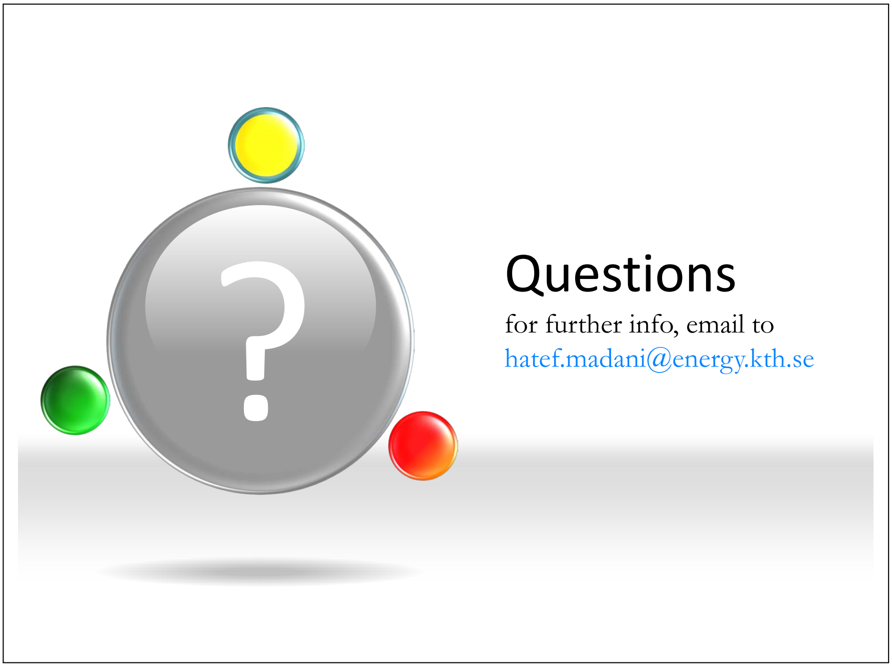
Publications

- Madani H., Claesson J., Lundqvist P. "A descriptive and comparative analysis of three common control techniques for an on/off controlled Ground Source Heat Pump (GSHP) system", submitted to International Journal of Energy and Buildings.

- Madani H., J. Acuna, B. Palm, J. Claesson, P. Lundqvist 2010 "The ground source heat pump: a system analysis with a particular focus on the U-pipe borehole heat exchanger" 14th ASME International Heat Transfer conference, Washington, US, ID IHTC-22395.

- Madani H., Wallin J., Claesson J., Lundqvist P. 2010 "Retrofitting a variable capacity heat pump to a ventilation heat recovery system: modeling and performance analysis", International Conference of Applied Energy, Singapore, ID 136.

- Wallin J., Madani H., Claesson J. 2012 "Run-around coil ventilation heat recovery system: A comparative study between different system configurations", International Journal of Applied Energy, Volume 90, Issue 1, pp 258-265.



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2.5 Energy Efficient Impulse Coolers

Per Henrik Pedersen *(prp@teknologisk.dk)*
DTI, Center for Refrigiation

Timetable ▲
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TECHNOLOGICAL
INSTITUTE

Agenda

1. Professional refrigerators and freezers
2. Regulation of energy efficiency and use of refrigerants in DK and the EU
3. The impulse cooler project
   - Background for the project
   - The existing cooler: How it works, lab tests and analysis
- Improvements:
  - Reducing air infiltration
  - Reducing other heat loads
  - Improvement of refrigeration system
  - Construction of 1st, 2nd and 3rd generation of prototypes. Lab test.
4. Discussion, conclusion and recommendations.

1. Professional refrigerators and freezers

Cabinet with integrated refrigeration system
Similar to domestic refrigerators, but:
- Tough use
- Tough requirements to temperatures
- Often bigger
- Higher energy consumption
- Often fans (inside and outside)

Two main types:
- Cabinets for sales purpose
- Cabinets for storage

1.04.06

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2. Regulation of energy efficiency and emission of greenhouse gases

Almost no regulation of energy efficiency!

- No Minimum Efficiency Performance Standard (MEPS)
- No Labeling
- But this will change!

Sales cabinets are often bought by big producers of soft drinks and ice-cream and installed in supermarkets. Energy efficiency has not been an issue until recently.

Major soft drink companies start asking for energy efficiency.

EU EcoDesign studies will (probably) result in MEPS and perhaps energy labeling:

- Sales cabinets: 2015 (?)
- Storage cabinets: 2014 (?)

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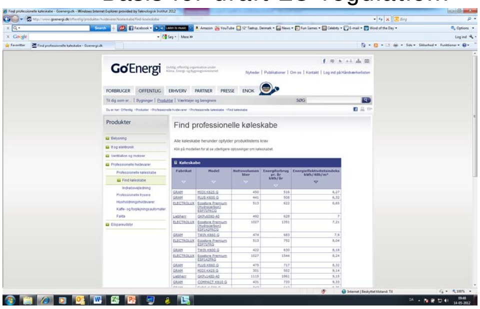
National schemes for storage cabinets: UK and DK

UK and DK have two almost identical schemes for professional storage cabinets.

Tough criteria: MEPS

1 door refrigerators:	15 kWh/48t/m^3
2 door refrigerators:	12 kWh/48t/m^3
1 door freezers:	40 kWh/48t/m^3
2 door freezers:	36 kWh/48t/m^3

Basis for draft EU-regulation!



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Special situation in Denmark (1)

Danish production of professional refrigerators and freezers:

Cabinets:
Vestfrost Solutions: Bottle coolers, wine coolers, impulse coolers, vaccine cooler

Gram Commercial: Professional kitchen refrigerators and freezers, blast chillers, pharmaceutical coolers

Elcold: Ice-cream cabinets, supermarket cabinets.

Components:
Danfoss: Components (control, expansion valves)

SECOP (Danfoss Compressors, Flensborg)

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Special situation in Denmark (2):

Regulation of F-gases (including HFCs):

Official Danish policy for promoting natural refrigerants:

1. Ban of HFCs: Use is banned (except in the interval of 0.15 – 10 kg)
2. Tax: 150 DKK/Tonnes CO_2-eq. (~200 DKK/kg HFC134a)
3. Support for alternatives: D-EPA has funded developing projects for developing alternative technology.

Special situation in Denmark (3):

Helped Danish manufacturers develop and market energy efficient appliances with natural refrigerants!

Vestfrost: First bottle cooler was marketed in 2001 (R600a)
Improved versions in field test in 2006 (Carlsberg, COOP, DTI)

Gram Commercial: First professional kitchen refrigerators and freezers marketed in 2002 (R290).
Now: becoming standard in Northern Europe.

Great success with these products!

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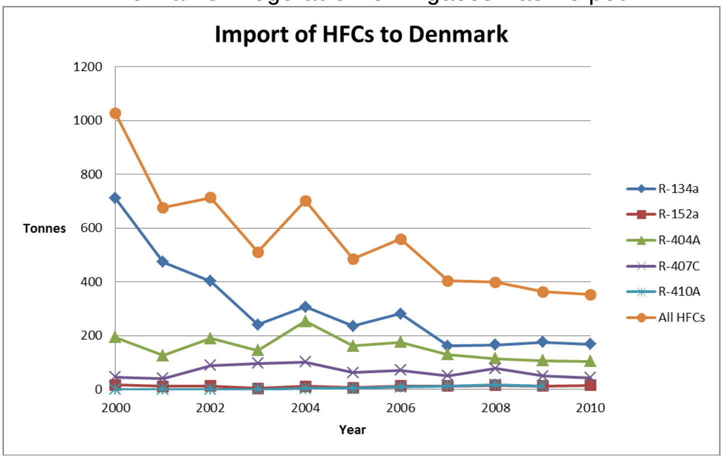
Special situation in Denmark (4):

The Danish regulation of F-gases has helped!



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3. The impulse cooler project
Background (1)

- Increased number of coolers (DK: 30.000?)
- High energy consumption (5 – 8 kWh/day or more)
- HFCs controlled by the Kyoto protocol, EU regulation and national regulation
- Strategic product for Vestfrost Solutions

- Economic support from “Dansk Energi” (ELFORSK) to develop an energy efficient concept for a new impulse cooler. The project was approved by the Danish Energy Agency.

Project partners:
Vestfrost Solutions A/S
IPU/DTU
COOP
PepsiCo
DTI (project manager)

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Background (2)

- Increasing attention on the electricity consumption of coolers placed in supermarkets, gas stations and kiosks.
- Declared policy of several producers of soft drinks to stop purchasing equipment with HFC refrigerants and instead purchase energy efficient equipment.
- There is an increasing market for small bottle coolers (impulse coolers) as well as a potential for improved energy efficiency.
- Restrictions and bans on HFC refrigerants due to their contribution to global warming.
- Vestfrost A/S wants to emphasise this business segment as they believe it will become a future strategic business area.

Background (3)

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- Vestfrost and DTI developed and tested large bottle coolers (400 litres with glass doors) with R600a (isobutane) and CO_2 as refrigerant and have good experience with both types of refrigerant.
- A field-test of 18 bottle coolers with R134a, R600a and CO_2 as refrigerant has taken place in co-operation with Carlsberg. The result shows that hydrocarbon bottle coolers are app. 28% more energy efficient compared to R134a. It also shows that CO_2 bottle coolers are app. 12% more energy efficient compared to R134a.
- In the light of the above-mentioned test, Carlsberg has decided to purchase bottle coolers based on hydrocarbons.

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Project phases

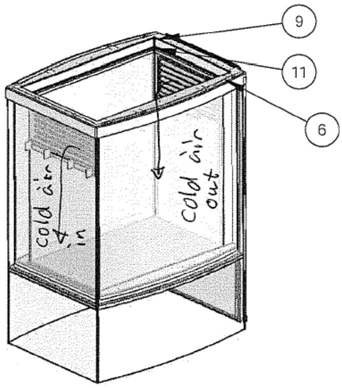
1. Collecting experience
2. Mathematical model
3. Possible solutions to reduce air infiltration
4. Possible solutions to improve efficiency of cooling system
5. Building prototypes
6. Testing prototypes
7. Evaluations
8. Reporting and presentation of results.

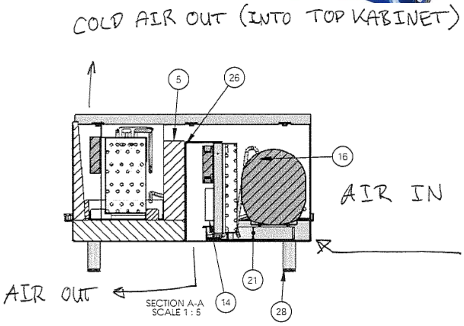
Aim of project:

- Reduce energy consumption (at least 25%)
- Using natural refrigerants
- New condenser – easy to clean
- Competitive product

The existing cooler

The existing cooler: Vestfrost POS 72R





Existing cooler (2)

- Compressor: Danfoss NL10MF
- R134a, Charge: ?
- 2 DC fans on evaporator
- 2 DC-fans on condenser
- Transformer (at condenser)
- Capillary tube
- Electronic thermostat, controlling temperature and defrosting
- Automatic evaporation of defrost water
- 4 diode lights at top of cooler

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Lab test at DTI

- Purpose: get an impression of how it works
- 25°C in climate chamber
- Test with and without lid

Without lid:
E = 3.99 kWh/24h (No 151)
E = 4.31 kWh/day (No 152)

Same level (or a little less than) other impulse coolers!

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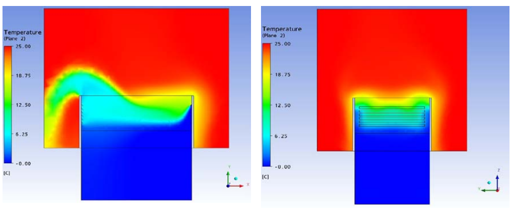
Models

Air infiltration, open lid:
Full cooler
25°C, 60% RH
CFD model

Q-Sides: 44 W
Q-ESP-foam: 6.4 W
Internal fans: 6.2 W
Lights: ?

Total: 251.6 W
(+ light + leakages
from warm to cold
in refrigeration system)

Infiltration	19%
infiltration heat load, total	195 W
Air cooling	80.5 W
Water cond.	101
Ice formation	13.5 W



Compare model and test results

- We estimated the evaporation temperature to –10°C and the condensation temperature to +55°C.
- The COP of the refrigeration system should be app. 1.39

- **Without lid:**
- Models predict the heat load to be app. 250 W
- Test shows electric consumption of app.167 W.
- App. 27 W is for fans, light, transformer and electronical thermostat.
- App. 140 W is for compressor. That gives a cooling capacity of 195 W, which is exactly as predicted by the model.

- Model:
- Q(without lid): 195 W
- Q(with lid): 30 W
- Reduction (with lid): 165 W

Discussion on improvements

Reducing air infiltration

- Better air curtain?
- Vestfrost idea: Removable lid
- Reduce leakage in refrigeration machine

Reducing other heat loads

- Better insulation

Improvement of cooling system

- Use state of the art R600a compressor: NLE15KTK.2 or smaller compressor
- 26% better COP
- Use of R290 compressors (NL7CN or TL5CN)?
- Improvement of condenser and evaporator?
- Improvement of air cooling and condenser?
- Improvement of cold air flow?

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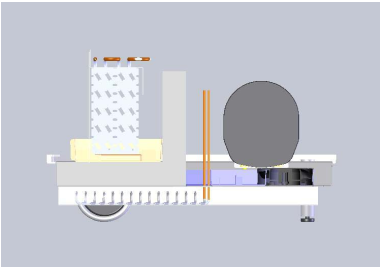
Construction and test of 1st generation of prototypes

The first generation of prototypes (one R134a + one R600a refrigerant), equipped with a new condenser type, made from smooth steel pipe which was easy to clean.

- Both prototypes worked badly! Bad heat transmission at condenser. Especially the R134a cooler was hot on the condenser side.
- Both coolers were running 100% of time.
- Shipped back to the Vestfrost Company.



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Second generation of prototype

Vestfrost strategic decision: Drop R134a and concentrate on R600a

Second generation impulse cooler: Better condenser

Result: The cooler "did the job" and performed in the same way as the original coolers. Energy consumption: 4.07 kWh/day.

Measuring points	Data sources		
Start		28-02-2009 13:19	01-03-2009 18:06
Stop		01-03-2009 13:19	02-03-2009 18:06
Duration		24:00 [HH.MM]	24:00 [HH.MM]
Ambient temperature			
Average temperature		24,9	24,8
Left	Z:\KFS\ANVILLE\03\01	25	24,9
Right	Z:\KFS\ANVILLE\03\02	24,7	24,7
Appliance			
Average temperature		2,1	2,1
Can 1	Z:\KFS\ANVILLE\03\03	0,5	0,5
Can 2	Z:\KFS\ANVILLE\03\04	2,9	2,8
Can 3	Z:\KFS\ANVILLE\03\05	0,8	0,8
Can 4	Z:\KFS\ANVILLE\03\06	2,9	2,9
Can 5	Z:\KFS\ANVILLE\03\07	1,2	1,2
Can 6	Z:\KFS\ANVILLE\03\08	1,8	1,8
Can 7	Z:\KFS\ANVILLE\03\09	3,6	3,6
Can 8	Z:\KFS\ANVILLE\03\10	2,2	2,2
Can 9	Z:\KFS\ANVILLE\03\11	3,2	3,2
Kompressor			
Voltage	Z:\KFS\PM100\12\VRMS	230,8	230
Power	Z:\KFS\PM100\12\WATT	170,7	169,3
Running time	Z:\KFS\PM100\12\WATT	100	100
Energy	Z:\KFS\PM100\12\WH	4072,7	4067,3
Energy consumption/24h	Z:\KFS\PM100\12\WH	4072,2	4066,9

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.. Second generation prototype (2)

Improvement of air curtain:
CFD programme and calculation was used to propose changes:

- Change prototype with additional air slots: only minor effect
- Change prototype with air slots + air guides: only minor effect.
- Change prototype with pulse control of evaporator fan during compressor off time (which was 27.5% of time): 8% decrease of energy consumption: not bad!

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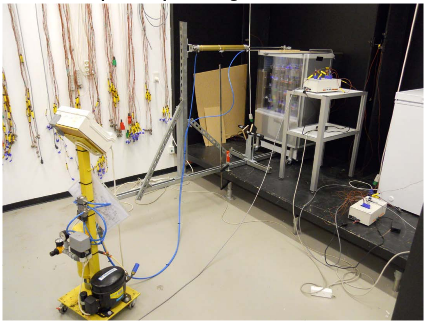
...second generation of prototype (3)

Automatic lid:
Test 11: Closed lid: Result: 2.393 kWh/day (about 40% saving)
Test 12: Lid openings after EN23953 (openings 12 hours per day, 6 openings/hour, 8 seconds per opening) – except first opening: 3 minutes. Result: 2.351 kWh/day
Test 13: as test 12, but 30 seconds per opening: Result: 2.771 kWh/day



That is quite good.

BUT:
PepsiCo didn't like the lid!
Wanted "open air cooler".
Task:
Develop third prototype!

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...second generation of prototype (4)

Turn off during night-time, open lid:

Test 15: Power shut off from 22.00 to 06.00: Energy consumption 2.606 kWh/day (minus 34.5% compared to original prototype and minus 29% compared to rebuilt prototype with pulse control).

Temperature of warmest can is 13.5°C at 10.00 am.

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Third generation prototype

Results from 1st and 2nd generation prototypes showed further potential for improving the condenser.

Third generation prototype:

- Box type condenser without fins (bigger surface and easy to clean)
- A new and slightly smaller compressor is used (Danfoss NLE9KTK)

Improvement inside the cabinet on cold side:

- A <u>construction change inside the cabinet</u>: prevents return air to mix up with air into the space between the side walls. That reduces the heat transmission.
- The air channel from the evaporator through the fans and into the cabinet has been changed and the <u>pressure drop has been decreased</u>. That brings more air into circulation which improves the air curtain (same type of fans).

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...third generation prototype (2)

Test 16: Test with open lid: E=1.907 kWh/day

Test 17. As test 16 but slightly colder thermostat setting: E= 2.022 kWh/day

Test 18: As test 17, but with closed lid. E= 1.805 kWh/day

The cooler has no LED light. To compare with previous prototypes 0.25 kWh/day was added to the result. The result is 2.215 kWh/day compared to 4.149 for original cooler (47% reduction).

Now lid closure only reduces energy consumption by 11%.

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5. Conclusion

Vestfrost Solutions has a big range of impulse coolers.
The project results have also been implemented in other products.
This has resulted in a change to natural refrigerants and new energy efficient products.
Natural refrigerants are standard (R600a) in all impulse coolers, and most customers are satisfied with this solution.
100 of the new cooler has been build and probes has been send to potential customers.
The production price is lower due to easier production.
Impulse coolers are important to Vestfrost.

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The project won the ELFORSK-prize 2011

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Discussion

Impulse coolers are installed all over the world where cold drinks are sold.
There is a big potential for energy savings.

Natural refrigerants can be used without additional costs.

But energy efficiency is not yet a competition parameter for impulse coolers:

- Impulse sales coolers are not yet subject to EU Ecodesign studies
- No Ecodesign criteria (MEPS) or labeling system can be expected in the near future (~3-4 yrs).
- Important that DK and others countries fight for Ecodesign measures for impulse coolers.

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2.6 Research and Development in Efficient Energy Use

Anders Stouge *(ast@danskenergi.dk)*
Danish Energy Association

Timetable ▲
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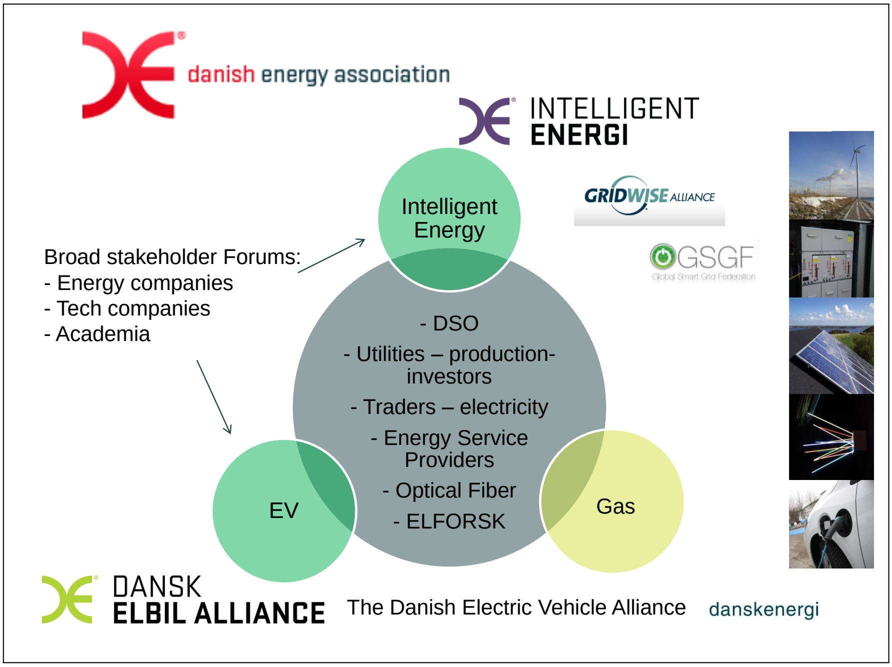
danish energy association
The Danish Energy Association is a commercial and professional organisation for Danish energy companies.
danishenergyassociation


danish energy association
INTELLIGENT ENERGI
Intelligent Energy
GRIDWISE ALLIANCE
GSGF
Broad stakeholder Forums:
- Energy companies
- Tech companies
- Academia
- DSO
- Utilities – production-investors
- Traders – electricity
- Energy Service Providers
- Optical Fiber
- ELFORSK
EV
Gas
DANSK ELBIL ALLIANCE
The Danish Electric Vehicle Alliance
danskenergi

New government – new goals – new agreement on energy

2020: 50% wind power in electricity consumption

2020: 40% reduction of GHG emissions vs. 1990

2030: Coal out of power plants

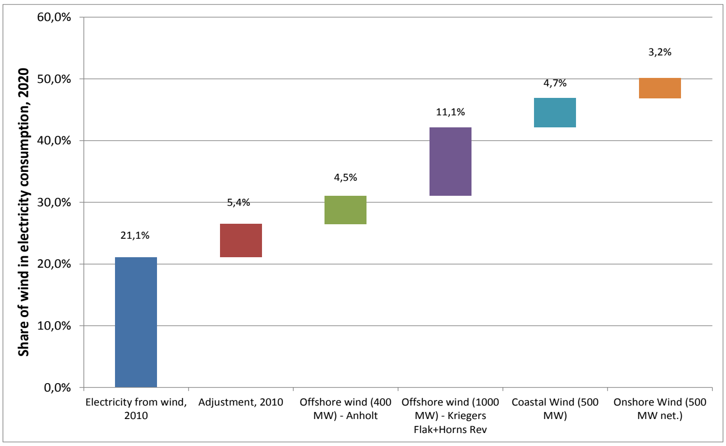
2035: 100% renewable energy in electricity and heating sector

2050: 100% renewable energy

“38-year-plan for making the Danish Energy System 100% renewable”

danishenergyassociation

50% wind power in electricity consumption
How?



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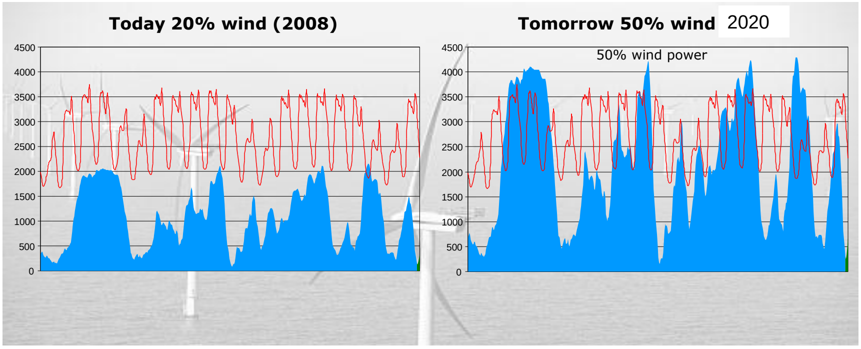
Calculations by Danish Energy Association

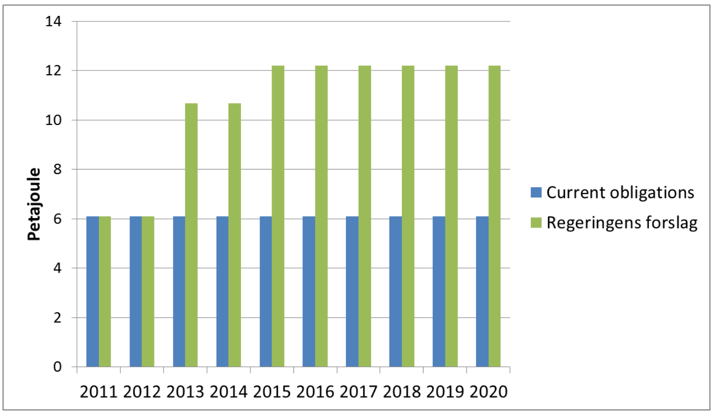
In future wind power will exceed demand in +1000 hours



Wind power covers the entire demand for electricity in 200 hours (West DK)

In the future wind power will exceed demand in more than 1,000 hours

Doubling Energy Saving Obligations for energy companies - Focus on buildings and industry



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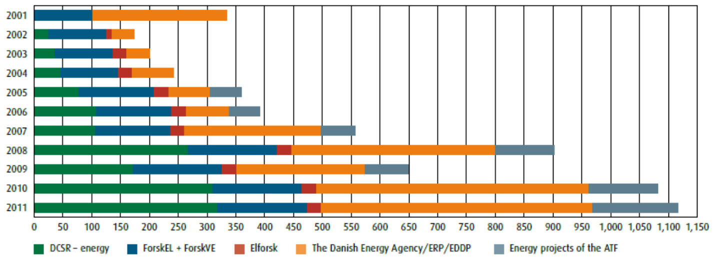
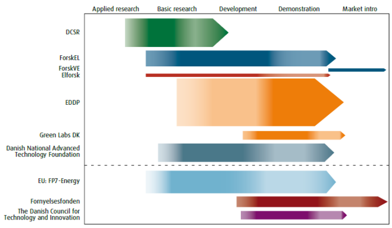
Funding for Energy Research - Public 2001-2011 (DKK mill.)





The Danish energy-plan will need new technologies, new knowledge ……

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Key priorities for ELFORSK

The energy companies' R&D programme for efficient and effective energy use is to fulfill the Danish vision:

- **to make end use energy consumption more efficient**
- **to convert fossil fuel use into efficient electricity use based on renewables**

- **to make the electricity consumption of the end user flexible through:**
  - End user "communication appliances" which can be controlled intelligently in connection with the demand of the electricity system – Smart Grid
  - Energy storage in the building stock and liquid for the running of Heat pumps and cooling systems

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From passive to active network management at DSO-level will be accompanied by developing new services for the electricity market

With active management of distribution networks, the amount of DG that can be connected to existing distribution networks ***can be increased by a factor of three to five without requiring network reinforcement!!!!.***

What's the Role for Heat Pumps????

Source: Akkermans and Gordijn,
Business Models for Distributed Energy Resources in a Liberalized Market Environment,
summarising report of BUSMOD, Enersearch AB, Malmö, Sweden, 2004.

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DSO - transition from passive to active network management –

In the active networks vision, the principles of network management differ from the classical view of networks

The 'infinite network' as customers used to know it, no longer exists!!!!!!.

The network interacts with its customers and is affected by whatever loads and generators are doing

A dynamic pricing system and a market for "using" the network at DSO-level will evolve – ***DSO will set the framework, standards and rules for the market***

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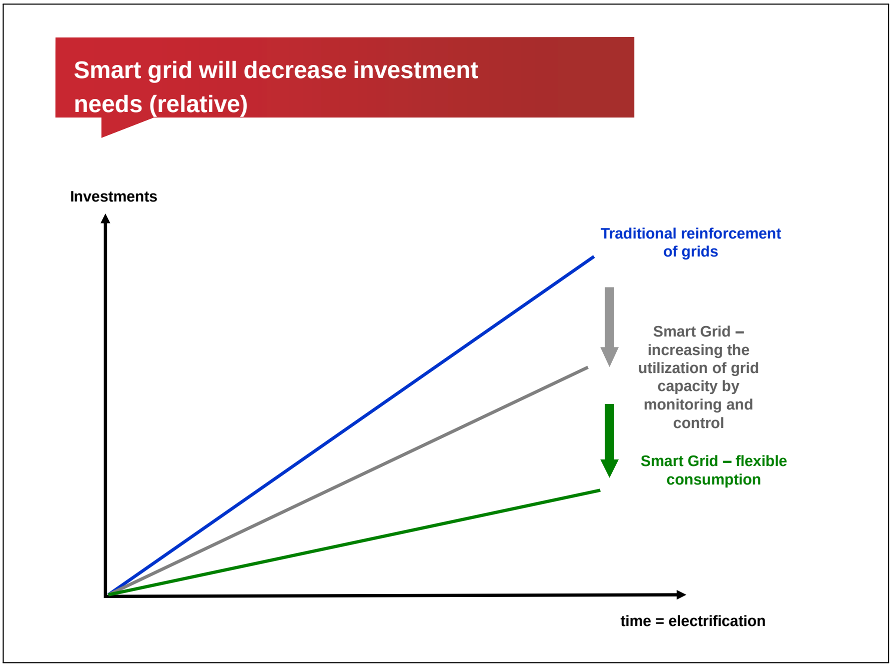
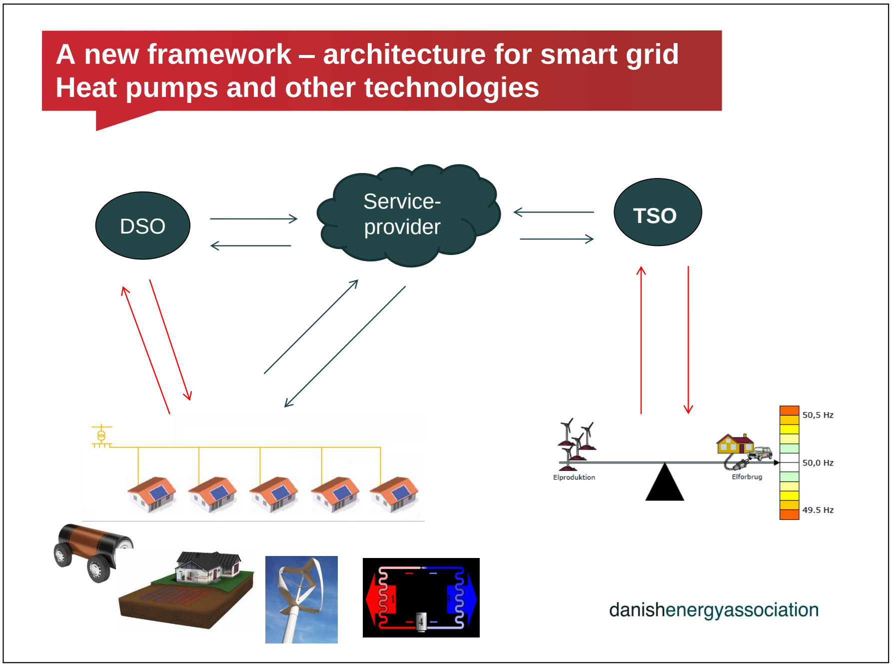

Smart grid will decrease investment needs (relative)
Investments
Traditional reinforcement of grids
Smart Grid – increasing the utilization of grid capacity by monitoring and control
Smart Grid – flexible consumption
time = electrification


A new framework – architecture for smart grid
Heat pumps and other technologies
DSO
Service-provider
TSO
Elproduktion
Elforbrug
50,5 Hz
50,0 Hz
49.5 Hz
danishenergyassociation


Introduce new technologies and market services for the optimization and utilization of grid capacity and to save energy
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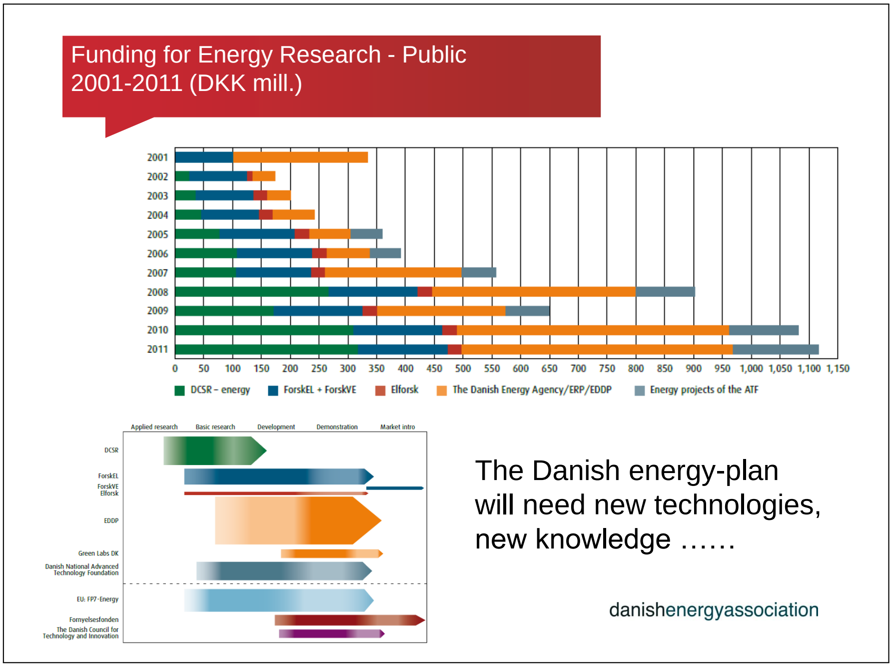
Funding for Energy Research - Public 2001-2011 (DKK mill.)
2001
2002
2003
2004
2005
2006
2007
2008
2009
2010
2011
0 50 100 150 200 250 300 350 400 450 500 550 600 650 700 750 800 850 900 950 1,000 1,050 1,100 1,150
DCSR – energy
ForskEL + ForskVE
Elforsk
The Danish Energy Agency/ERP/EDDP
Energy projects of the ATF
Applied research
Basic research
Development
Demonstration
Market intro
DCSR
ForskEL
ForskVE
Elforsk
EDDP
Green Labs DK
Danish National Advanced Technology Foundation
EU: FP7-Energy
Fornyelsesfonden
The Danish Council for Technology and Innovation
The Danish energy-plan will need new technologies, new knowledge ……
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Focus areas for cooling and heat pump systems for flexible use - ELFORSK

- Flexible solutions with integrated storage capacity
- Design of flexible high-temperature heat pumps utilizing waste heat from industry
- Intelligent controlling of airflow and adjustable set points of temperature using cooling and heat pump systems which will lead to reductions in energy consumption and flexibility
- Interaction between current electricity consumption in industrial processes and the electricity system's demand for consumers as suppliers of regulating power services

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Examples of finished and ongoing projects on cooling and heat pump systems

- Simulation tools for analyzing the possibilities for savings and flexibility in cooling systems with carbon dioxide as cooling agent
- Future cooling towers – utilization combined with dry coolers
- Efficient high-temperature heat pumps for industrial cooling
- Heat pumps equipped with vertical drillings as heat source utilizing a storage tank
- Aquifer Thermal Energy Storage - ATES
- Tools for analyzing, energy efficiency, and flexible hybrid systems for district cooling and heating
- Energy efficient milk cooling with intelligent control

- Find projects at "energiforskning.dk"

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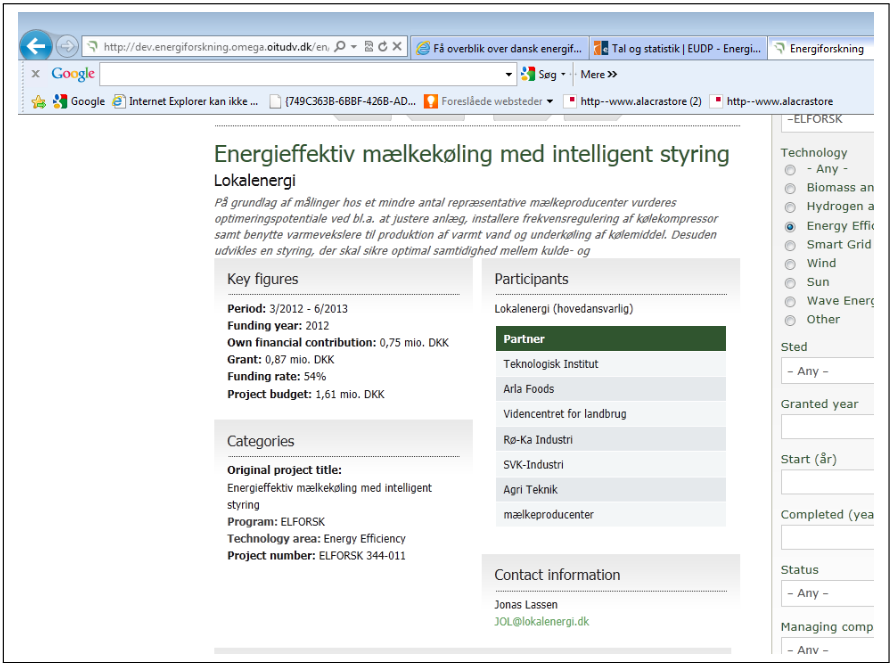


Energieffektiv mælkekøling med intelligent styring

Lokalenergi

På grundlag af målinger hos et mindre antal repræsentative mælkeproducenter vurderes optimeringspotentiale ved bl.a. at justere anlæg, installere frekvensregulering af kølekompressor samt benytte varmevekslere til produktion af varmt vand og underkøling af kølemiddel. Desuden udvikles en styring, der skal sikre optimal samtidighed mellem kulde- og

Key figures

Period: 3/2012 - 6/2013
Funding year: 2012
Own financial contribution: 0,75 mio. DKK
Grant: 0,87 mio. DKK
Funding rate: 54%
Project budget: 1,61 mio. DKK

Categories

Original project title: Energieffektiv mælkekøling med intelligent styring
Program: ELFORSK
Technology area: Energy Efficiency
Project number: ELFORSK 344-011

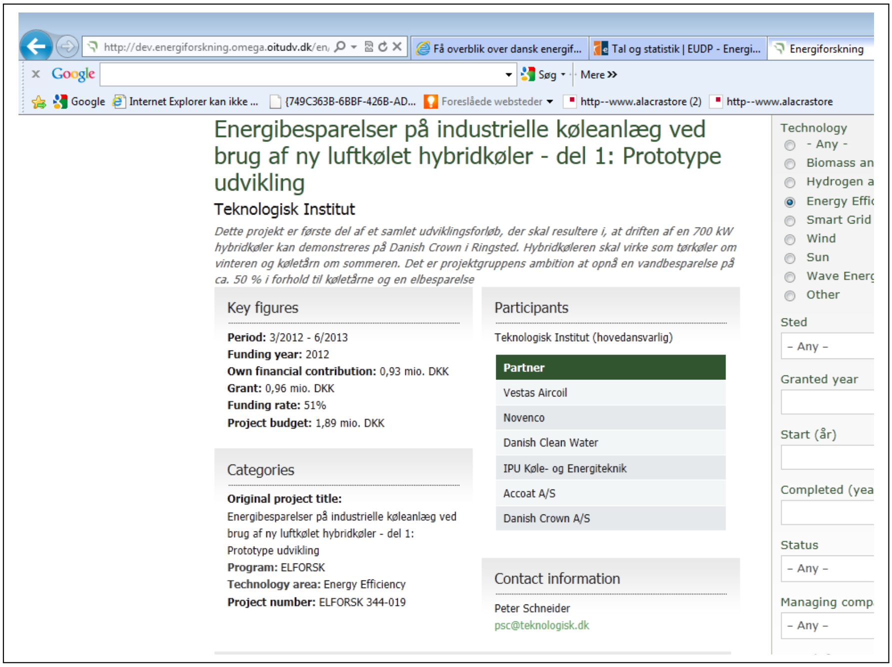
Participants

Lokalenergi (hovedansvarlig)

Partner
Teknologisk Institut
Arla Foods
Videncentret for landbrug
Rø-Ka Industri
SVK-Industri
Agri Teknik
mælkeproducenter

Contact information

Jonas Lassen
JOL@lokalenergi.dk



Energibesparelser på industrielle køleanlæg ved brug af ny luftkølet hybridkøler - del 1: Prototype udvikling

Teknologisk Institut

Dette projekt er første del af et samlet udviklingsforløb, der skal resultere i, at driften af en 700 kW hybridkøler kan demonstreres på Danish Crown i Ringsted. Hybridkøleren skal virke som tørkøler om vinteren og køletårn om sommeren. Det er projektgruppens ambition at opnå en vandbesparelse på ca. 50 % i forhold til køletårne og en elbesparelse

Key figures

Period: 3/2012 - 6/2013
Funding year: 2012
Own financial contribution: 0,93 mio. DKK
Grant: 0,96 mio. DKK
Funding rate: 51%
Project budget: 1,89 mio. DKK

Categories

Original project title: Energibesparelser på industrielle køleanlæg ved brug af ny luftkølet hybridkøler - del 1: Prototype udvikling
Program: ELFORSK
Technology area: Energy Efficiency
Project number: ELFORSK 344-019

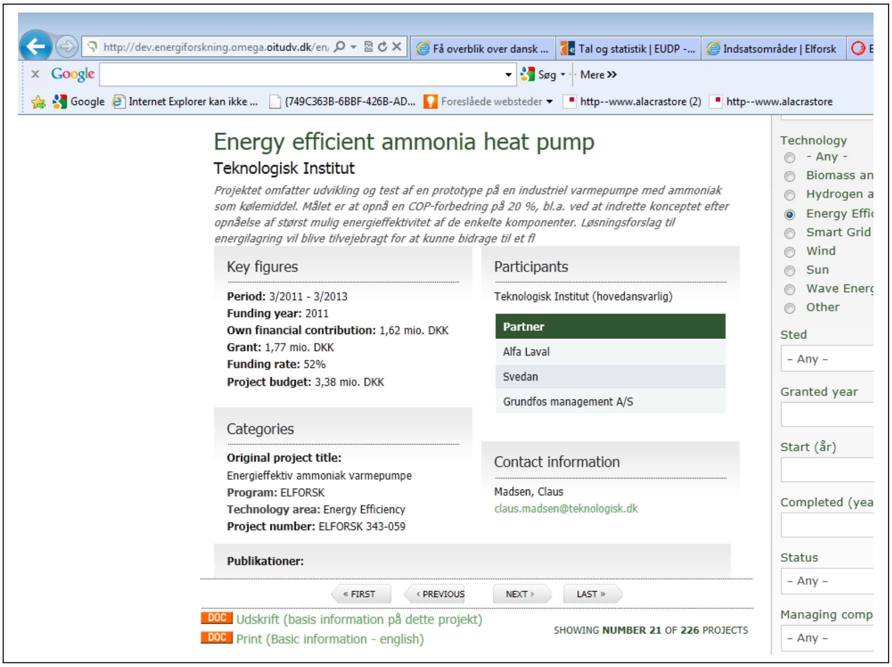
Participants

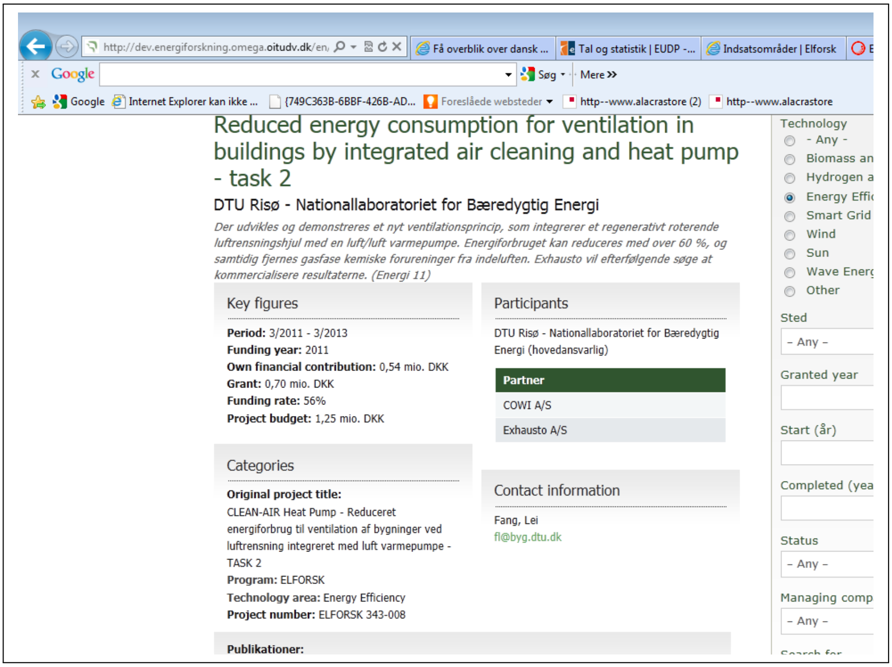
Teknologisk Institut (hovedansvarlig)

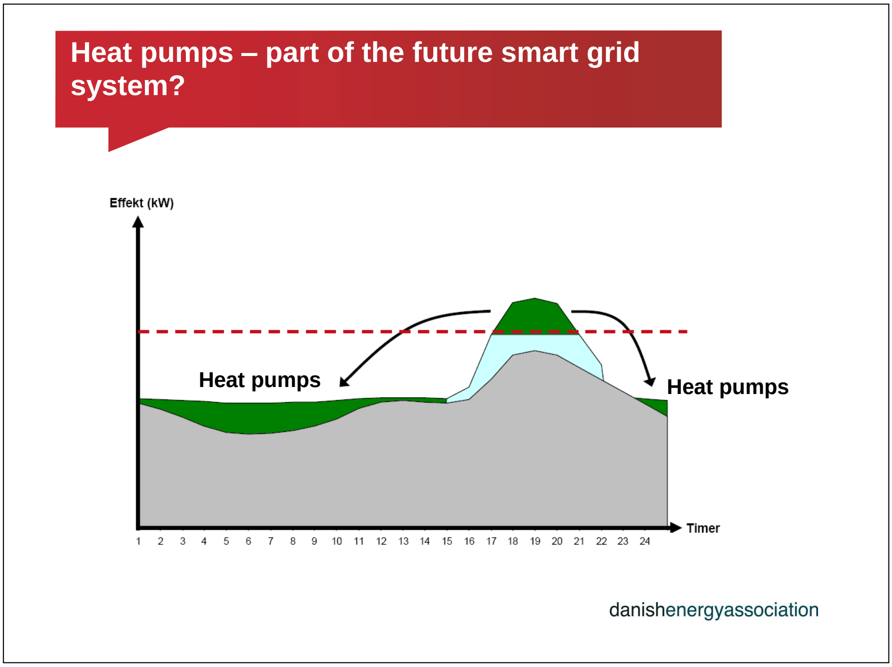
Partner
Vestas Aircoil
Novenco
Danish Clean Water
IPU Køle- og Energiteknik
Accoat A/S
Danish Crown A/S

Contact information

Peter Schneider
psc@teknologisk.dk


http://dev.energiforskning.omega.oitudv.dk/en/
Få overblik over dansk ...
Tal og statistik | EUDP -...
Indsatsområder | Elforsk
Google
Søg
Mere »
Google
Internet Explorer kan ikke ...
{749C363B-6BBF-426B-AD...
Foreslåede websteder
http--www.alacrastore (2)
http--www.alacrastore
Energy efficient ammonia heat pump
Teknologisk Institut
Projektet omfatter udvikling og test af en prototype på en industriel varmepumpe med ammoniak som kølemiddel. Målet er at opnå en COP-forbedring på 20 %, bl.a. ved at indrette konceptet efter opnåelse af størst mulig energieffektivitet af de enkelte komponenter. Løsningsforslag til energilagring vil blive tilvejebragt for at kunne bidrage til et fl
Key figures
Period: 3/2011 - 3/2013
Funding year: 2011
Own financial contribution: 1,62 mio. DKK
Grant: 1,77 mio. DKK
Funding rate: 52%
Project budget: 3,38 mio. DKK
Participants
Teknologisk Institut (hovedansvarlig)
Partner
Alfa Laval
Svedan
Grundfos management A/S
Categories
Original project title:
Energieffektiv ammoniak varmepumpe
Program: ELFORSK
Technology area: Energy Efficiency
Project number: ELFORSK 343-059
Contact information
Madsen, Claus
claus.madsen@teknologisk.dk
Publikationer:
« FIRST
‹ PREVIOUS
NEXT ›
LAST »
DOC Udskrift (basis information på dette projekt)
DOC Print (Basic information - english)
SHOWING NUMBER 21 OF 226 PROJECTS
Technology
- Any -
Biomass an
Hydrogen a
Energy Effic
Smart Grid
Wind
Sun
Wave Energ
Other
Sted
- Any -
Granted year
Start (år)
Completed (yea
Status
- Any -
Managing comp
- Any -


http://dev.energiforskning.omega.oitudv.dk/en/
Få overblik over dansk ...
Tal og statistik | EUDP -...
Indsatsområder | Elforsk
Google
Søg
Mere »
Google
Internet Explorer kan ikke ...
{749C363B-6BBF-426B-AD...
Foreslåede websteder
http--www.alacrastore (2)
http--www.alacrastore
Reduced energy consumption for ventilation in buildings by integrated air cleaning and heat pump - task 2
DTU Risø - Nationallaboratoriet for Bæredygtig Energi
Der udvikles og demonstreres et nyt ventilationsprincip, som integrerer et regenerativt roterende luftrensningshjul med en luft/luft varmepumpe. Energiforbruget kan reduceres med over 60 %, og samtidig fjernes gasfase kemiske forureninger fra indeluften. Exhausto vil efterfølgende søge at kommercialisere resultaterne. (Energi 11)
Key figures
Period: 3/2011 - 3/2013
Funding year: 2011
Own financial contribution: 0,54 mio. DKK
Grant: 0,70 mio. DKK
Funding rate: 56%
Project budget: 1,25 mio. DKK
Participants
DTU Risø - Nationallaboratoriet for Bæredygtig Energi (hovedansvarlig)
Partner
COWI A/S
Exhausto A/S
Categories
Original project title:
CLEAN-AIR Heat Pump - Reduceret energiforbrug til ventilation af bygninger ved luftrensning integreret med luft varmepumpe - TASK 2
Program: ELFORSK
Technology area: Energy Efficiency
Project number: ELFORSK 343-008
Contact information
Fang, Lei
fl@byg.dtu.dk
Publikationer:
Technology
- Any -
Biomass an
Hydrogen a
Energy Effic
Smart Grid
Wind
Sun
Wave Energ
Other
Sted
- Any -
Granted year
Start (år)
Completed (yea
Status
- Any -
Managing comp
- Any -



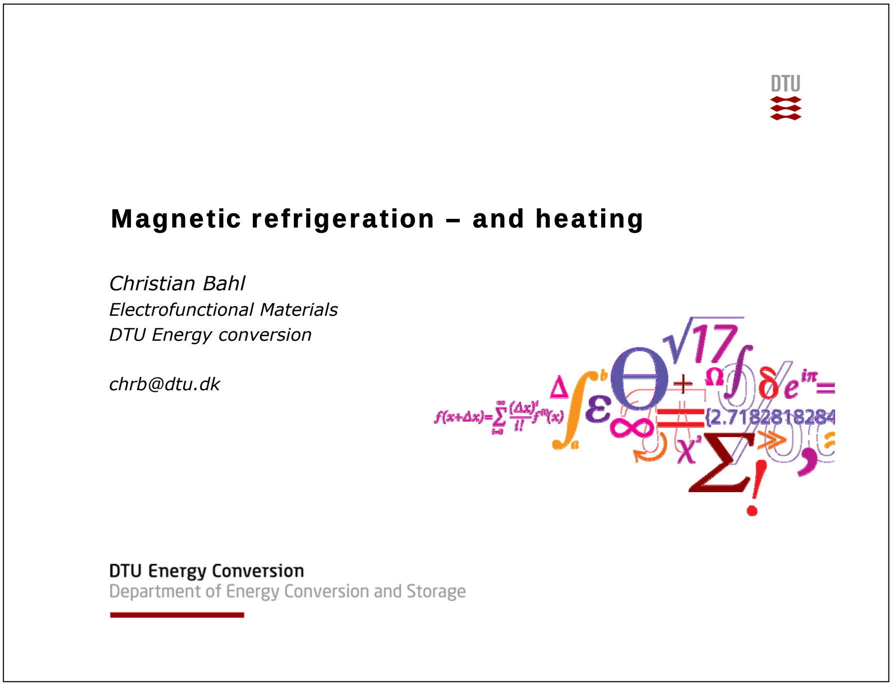
Go back to the table of contents ⊼ or to the timetable ▲

3 Presentations

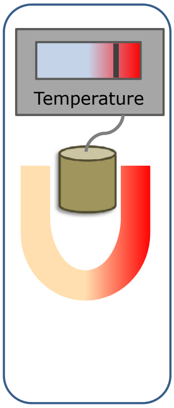
3.1 Magnetic Refrigeration – and Heating

Christian Bahl *(chrb@dtu.dk)*
DTU Energy Conversion

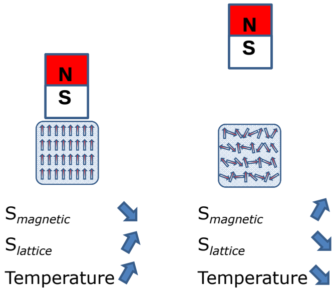
Timetable ▲
Table of contents



The magnetocaloric effect



- The magnetocaloric effect is reversible – efficient
- Solids material – no harmful gasses



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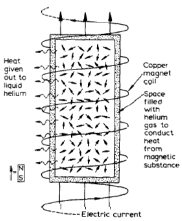
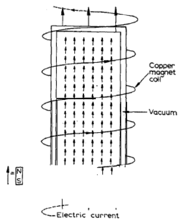
Low temperature Magnetic refrigeration

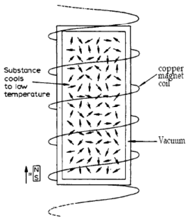
- Pioneered independently by Debye and Giauque in 1926/1927

  Nobel Prize 1949 

- "One-shot" process - need for a cyclic process







Giauque, Nobel lecture 1949

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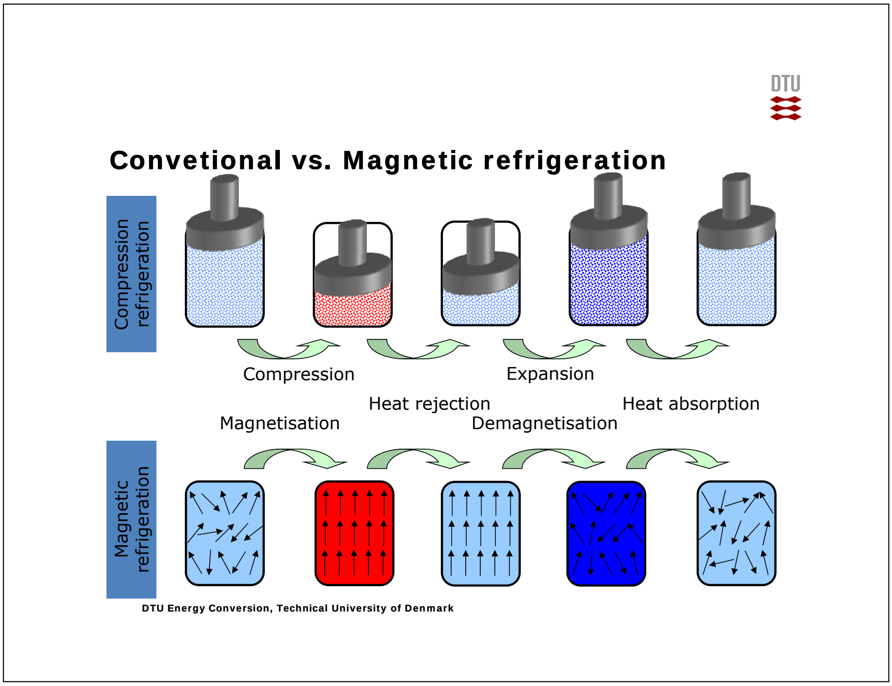
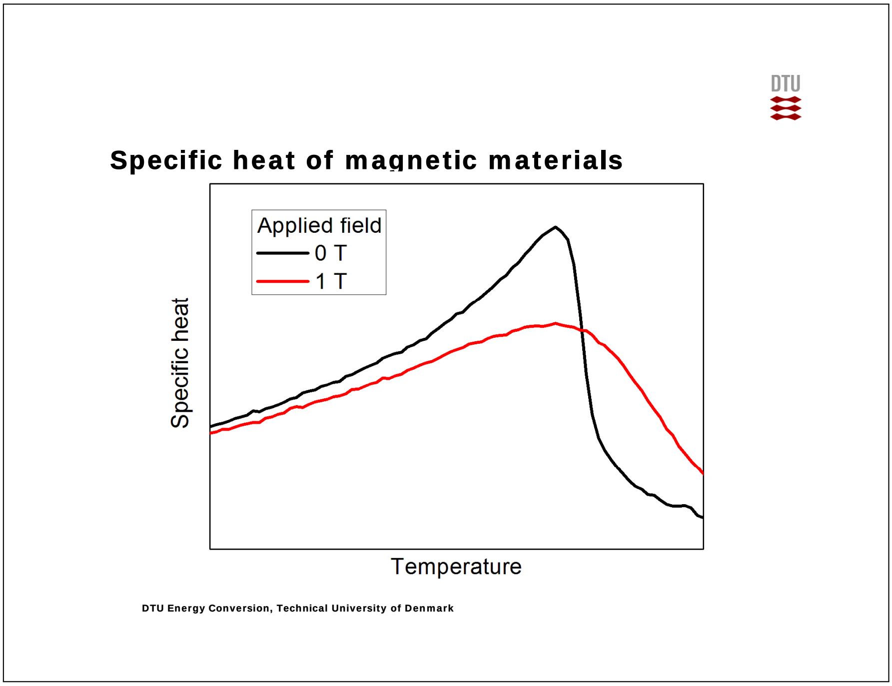
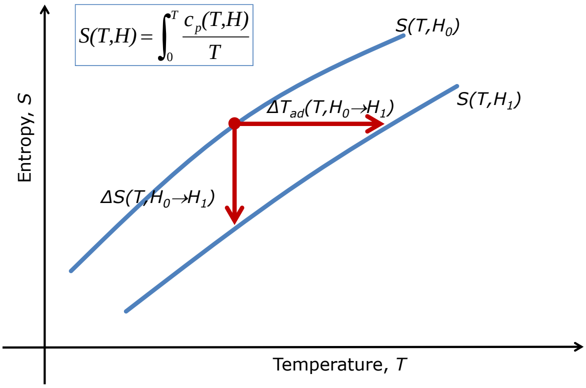

DTU
Convetional vs. Magnetic refrigeration
Compression refrigeration
Compression
Expansion
Heat rejection
Heat absorption
Magnetisation
Demagnetisation
Magnetic refrigeration
DTU Energy Conversion, Technical University of Denmark


DTU
Specific heat of magnetic materials
Applied field
0 T
1 T
Specific heat
Temperature
DTU Energy Conversion, Technical University of Denmark

The magnetocaloric effect

$$S(T,H) = \int_0^T \frac{c_p(T,H)}{T}$$



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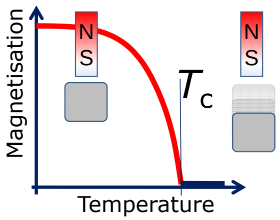
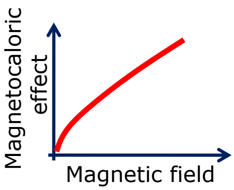
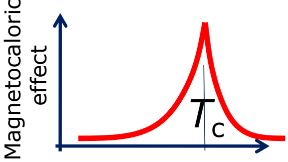
Characteristics of the magnetocaloric effect

$$\Delta S_M = \mu_0 \int_{H_1}^{H_2} \left(\frac{\partial M}{\partial T}\right)_H dH$$

$$\Delta T_{ad} = -\mu_0 \int_{H_1}^{H_2} \frac{T}{C_{P,H}} \left(\frac{\partial M}{\partial T}\right)_H dH$$







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Curie-temperature

- The following ferromagnetic elements are found in nature

Material	Curie Temperature
Cobalt (Co)	1115 ºC
Iron (Fe)	770 ºC
Nickel (Ni)	354 ºC
Gadolinium (Gd)	20 ºC
Terbium (Tb)	-53 ºC
Dysprosium (Dy)	-185 ºC

- Additionally there are many alloys, oxides etc.

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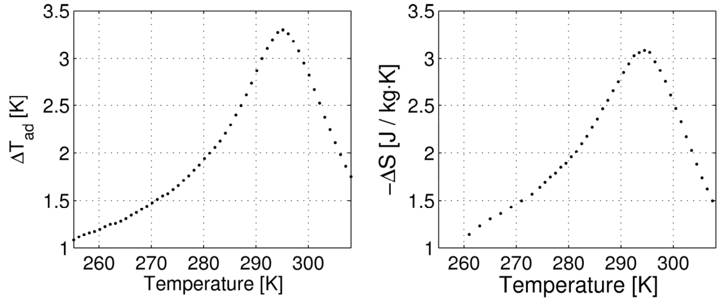
Gadolinium metal

- Curie temperature around 20° C
- Magnetocaloric properties in an applied field of 1 T.



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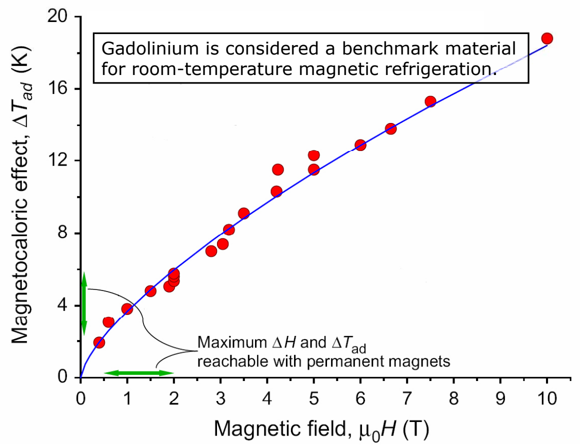
The magnetocaloric effect is small !



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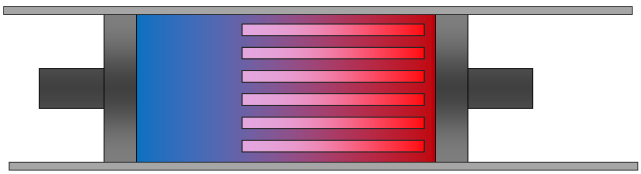
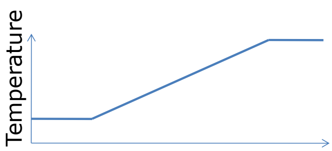
Pecharsky & Gschneidner, Int. J. Refr., 2006

Active Magnetic Regeneration





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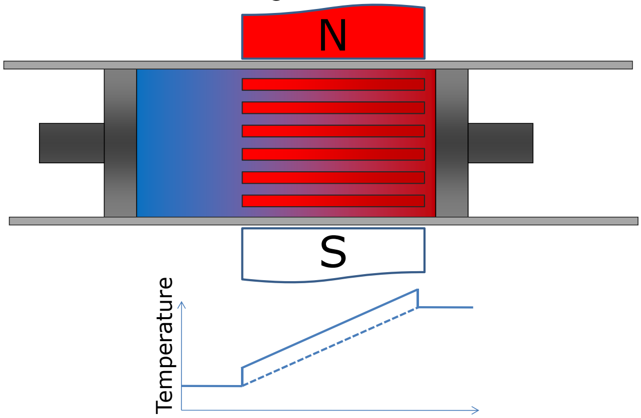
The material is magnetised



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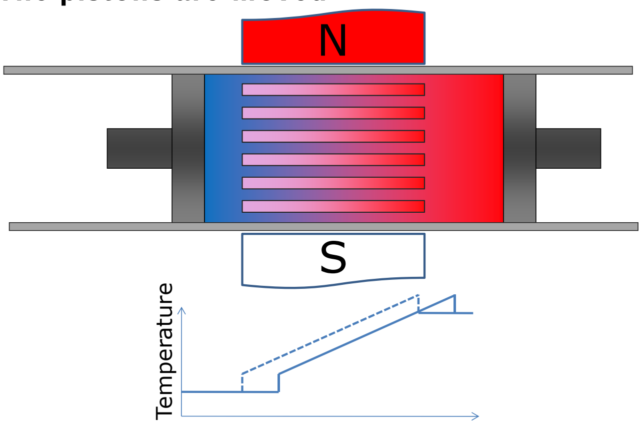
The pistons are moved



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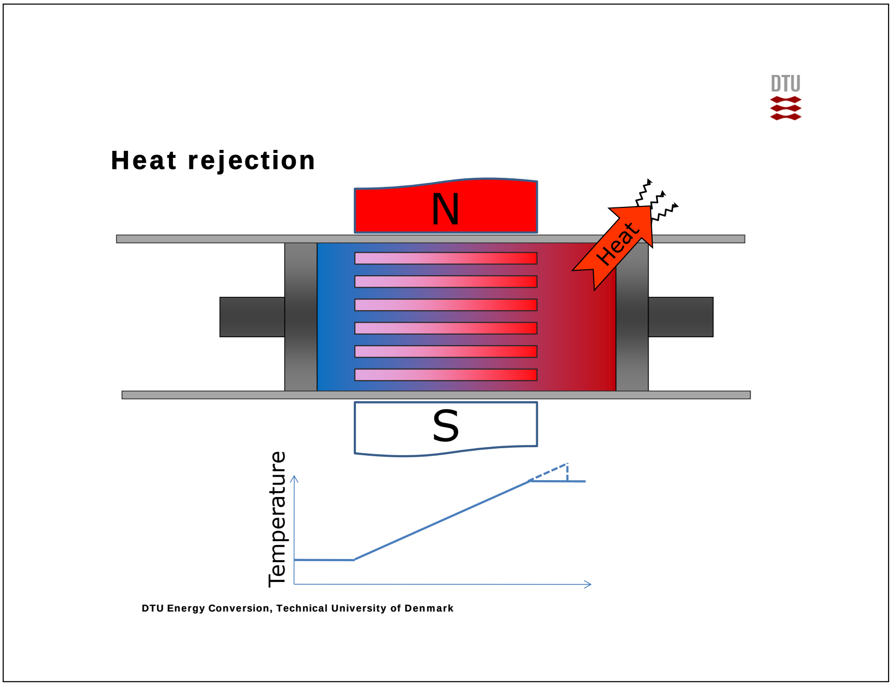
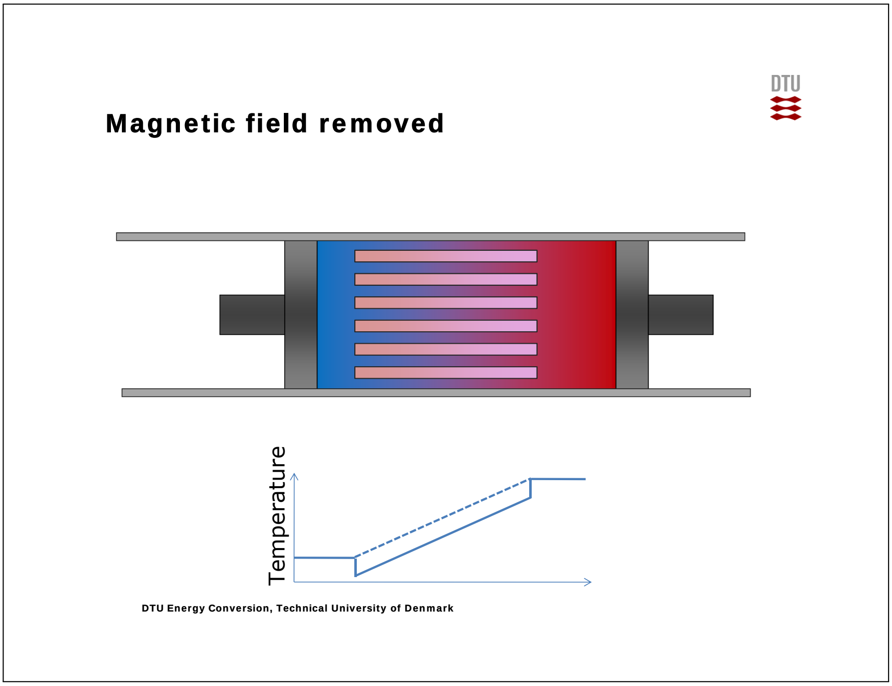
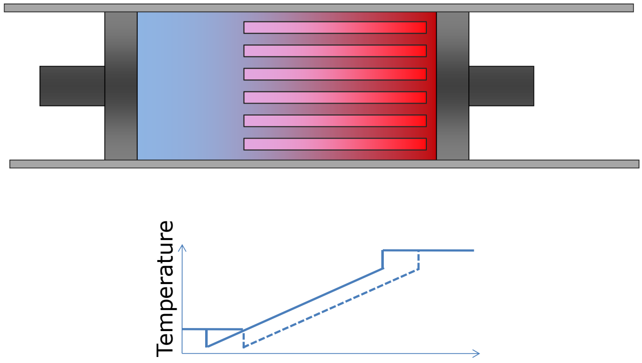

Heat rejection
N
Heat
S
Temperature
DTU Energy Conversion, Technical University of Denmark


Magnetic field removed
Temperature
DTU Energy Conversion, Technical University of Denmark

Pistons moved back



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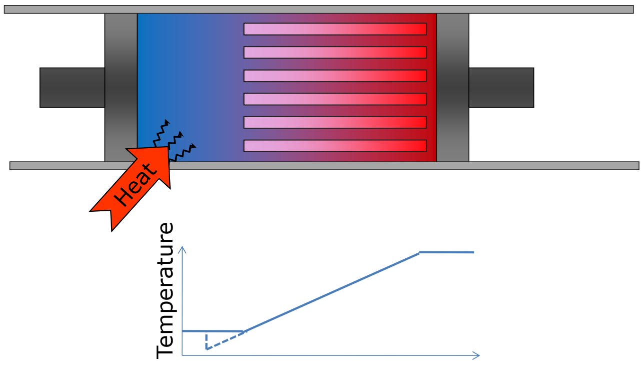
Heat absorbtion



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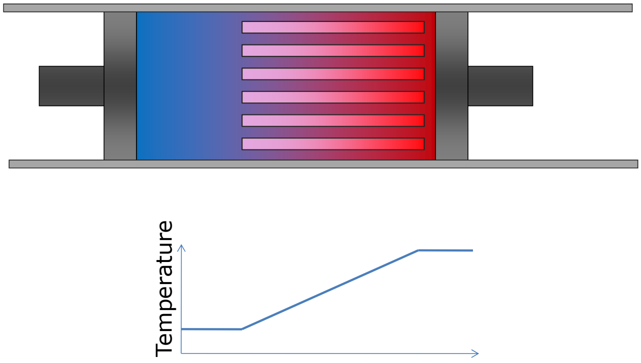
Back to start



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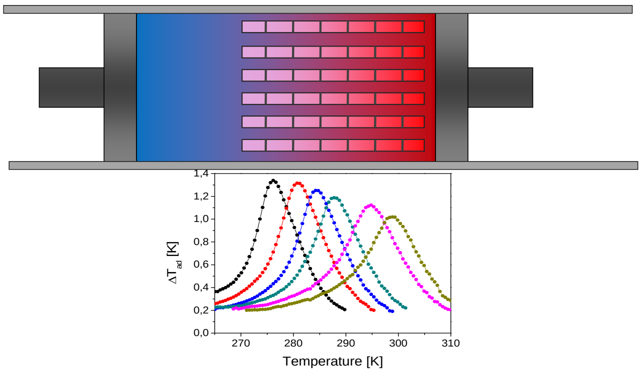
Optimising the Curie temp.



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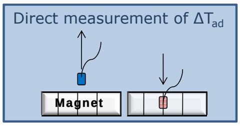
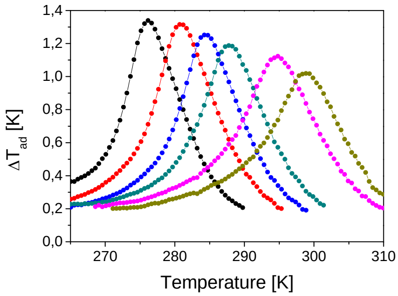
Perovskite manganites

- A series of manganites has been prepared.
- The Curie temperature can be controlled.
- It is cheap to process into the desired structures.

$La_{0.67}Ca_{0.33-x}Sr_xMn_{1.05}O_3$
x = 0.04 – 0.09





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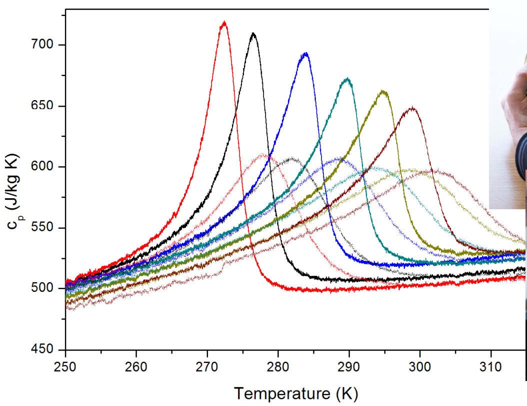
Specific heat in zero and 1 T fields



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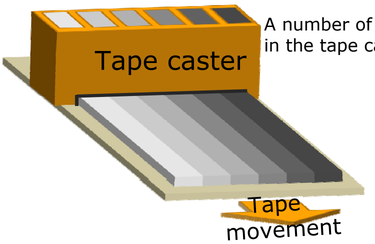
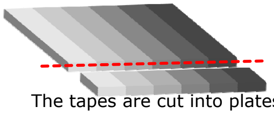
Multi-material adjacent tape casting



A number of ceramic slurries are loaded in the tape caster.



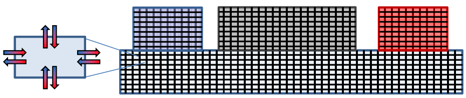
The tapes are cut into plates

Actual 10 material ceramic plate

DTU Energy Conversion, Technical University of Denmark

Numerical model of magnetic refrigeration

- Advanced 2D model combining:
    - o Heat transfer
    - o Fluid flow
    - o Magnetic properties

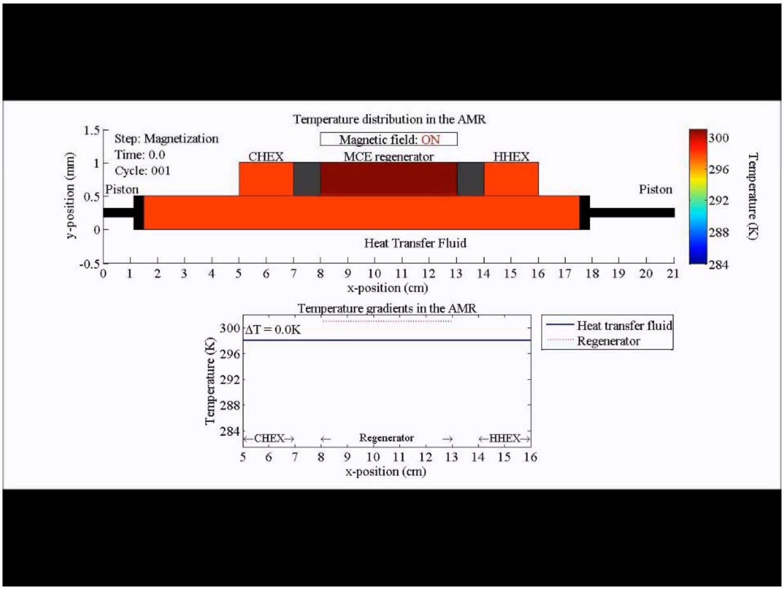


- The model is used to calculate performance of magnetic refrigeration.

- The advantage of using multiple materials can be studied.

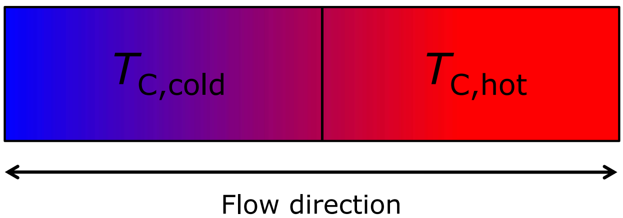
K.K. Nielsen et al., Int. J. Refrig. **33**, 753-764 (2010)
K.K. Nielsen et al., Int. J. Refrig. **32**, 1478-1486 (2009)
K.K. Nielsen, Ph.D. Thesis, 2010, Risø DTU

DTU Energy Conversion, Technical University of Denmark



2-material AMR concept

Cold side (inside the refrigerator) Hot side (ambient)

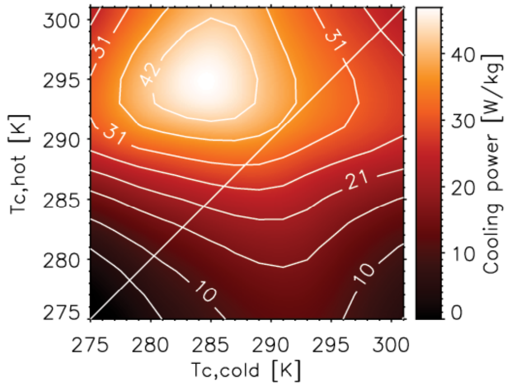


Assessment parameters:

Cooling power at a fixed temperature span: Q_c [W/kg]

DTU Energy Conversion, Technical University of Denmark

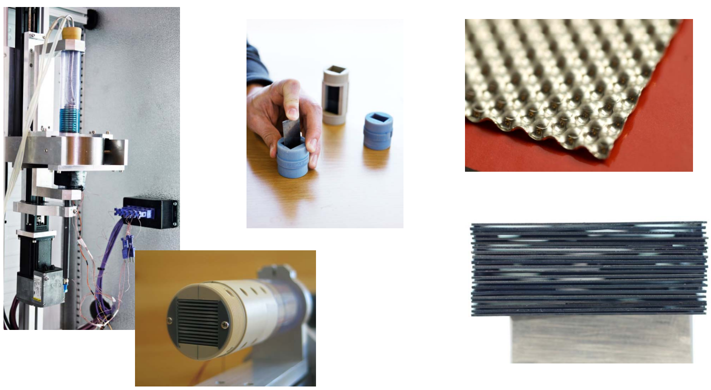
2-material, Gd-like AMR: cooling power @$\Delta T = 20\,\mathrm{K}$



Nielsen et al. 2010, Thermag IV proceedings

DTU Energy Conversion, Technical University of Denmark

Experiments in a "Test device"



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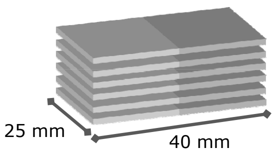
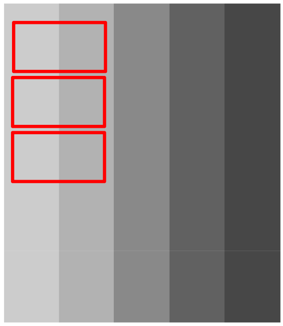
Testing multimaterials

- Plates with two materials cut from 5 stripe tapes.
- Size is chosen according to test machine:
  - Stack of 28 plates (40 mm x 25 mm) and a total mass of 51g.
  - 0.3 mm plates are stacked with 0.3 mm fluid channels.





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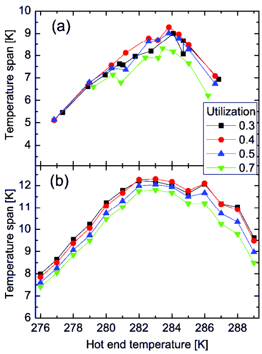
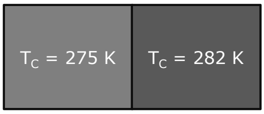
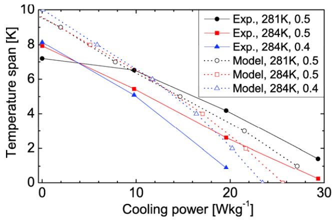
2-material LCSM plate experiments



Highest value with 1 mm Gd plates is 10.1 K.
The best result with a single material is 6 K.





Highest value obtained with 1 mm Gd plates is 16 W/kg.

Bahl et al. APL (In press)

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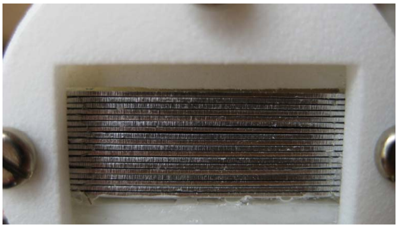
Assessing the impact of stacking quality of parallel-plate regenerators

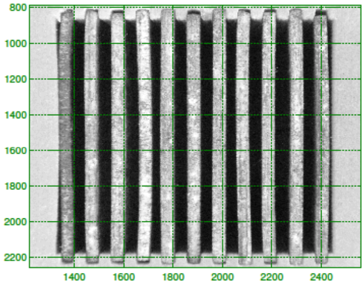
- Previous modelling has assumed perfectly spaced plates



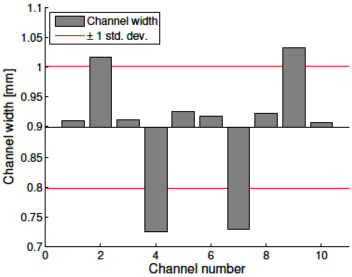
J.B. Jensen *et al*. Int. J. Heat and Mass Transf. **53** (2010) 5065

DTU Energy Conversion, Technical University of Denmark

Distribution of plate spacings in AMR



Image



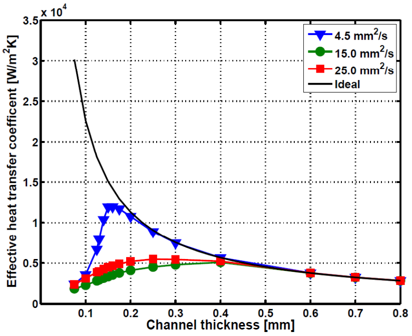
Measurements

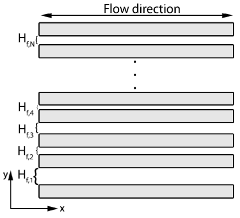
J.B. Jensen, PhD thesis, DTU 2011

DTU Energy Conversion, Technical University of Denmark

Maldistribution of channels

- The channel thickness is modelled from 0.08 mm to 0.8 mm keeping the standard deviation fixed at 0.035 mm.





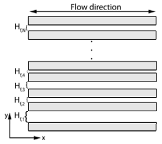
K.K. Nielsen *et al.* Appl. Therm. Engineering (Accepted)

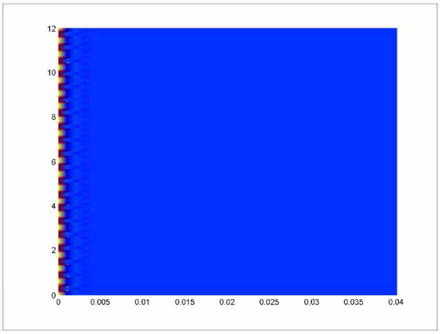
DTU Energy Conversion, Technical University of Denmark

Maldistribution of flow – an example

- Hot water is pushed between the cold plates
- Number of channels/plates: 20
- Mean channel thickness: 0.2 mm
- Standard deviation: 10 %
- Channel fluid: Water

- Plate thickness: 0.4 mm
- Plate material: Aluminium





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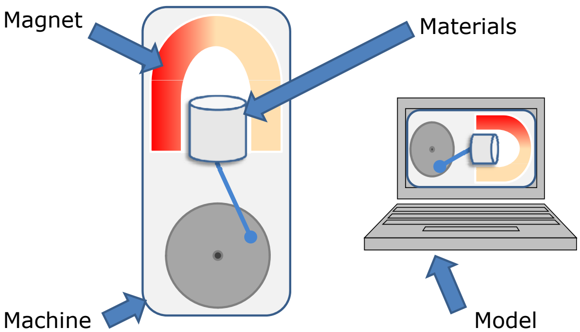
DTU

Designing magnetic refrigeration devices



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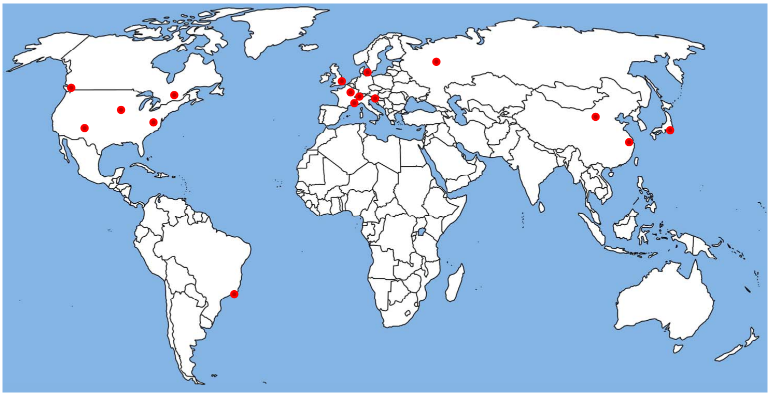
Magnetic refrigeration devices in the world



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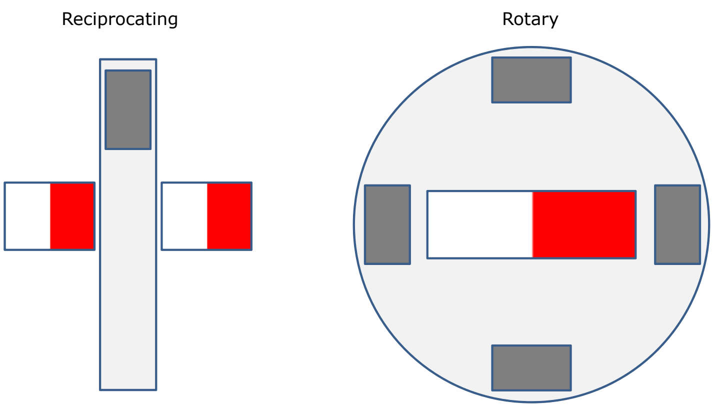
Magnetic refrigeration devices in the world

DTU Energy Conversion, Technical University of Denmark

Two types of magnetocaloric devices



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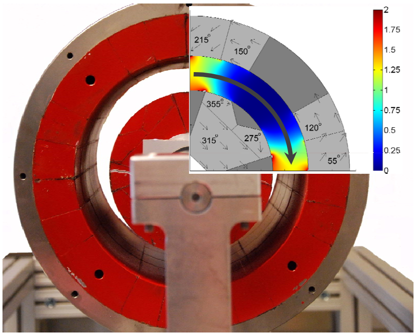
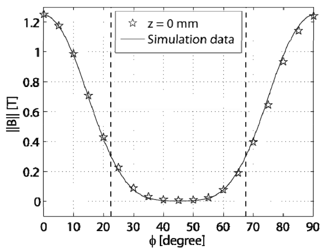
Design of a permanent magnet field source

- Good correspondence between numerical model and measured values.





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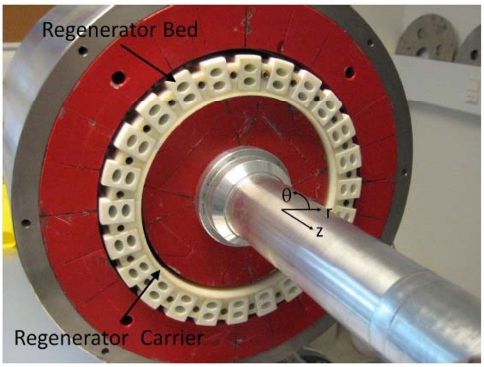
The rotating compartments in the magnet



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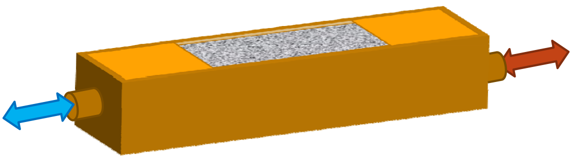
Cassette with Gd spheres (~0.5 mm)

- 120 g Gd in each of the 24 cassettes.
- Water can be pushed back and forth.



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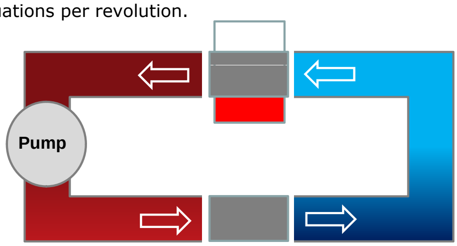
The flow system

- Each bed experiences an alternating flow.
- 8 flow situations per revolution.



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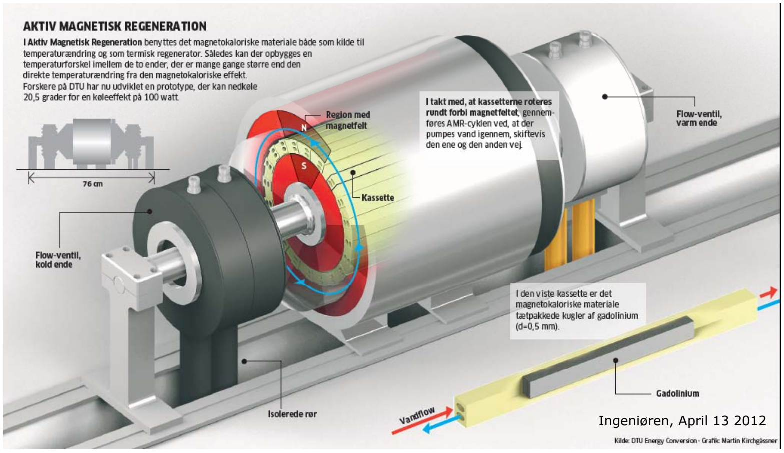
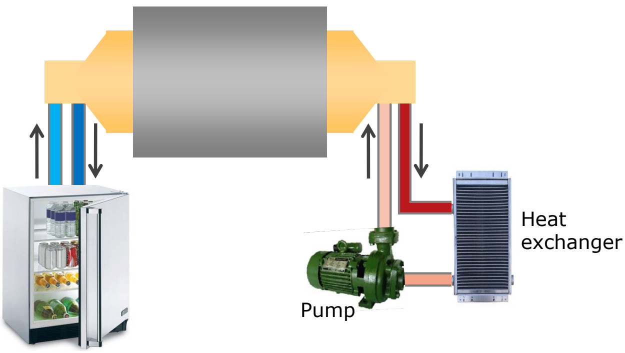
Prototype device



Ingeniøren, April 13 2012

Flow circuit

- The flow circuit includes pump, heat exchanger and heat load.



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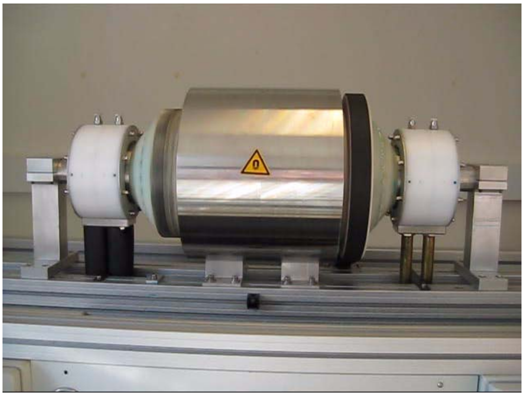
The prototype machine in action



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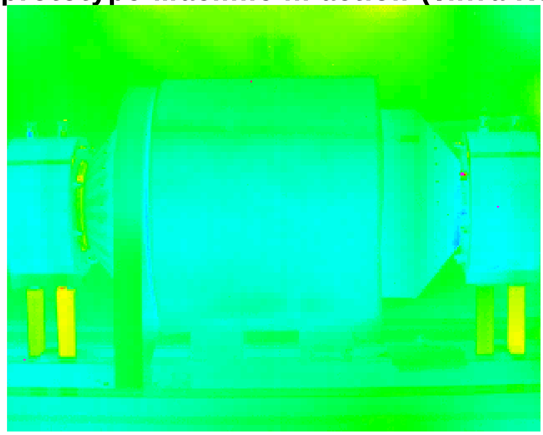
The prototype machine in action (Infra Red)



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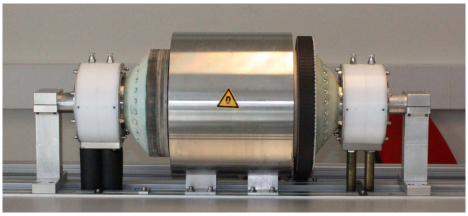
Best results to date

- Maximum load of **1010 W at zero temperature span**
- Temperature span of **25.4 K at no-load**
- Maximum frequency **10 Hz**

- Temperature span of **20.5 K at 100 W**
- Temperature span of **18.9 K at 200 W**
- Temperature span of **13.8 K at 400 W**



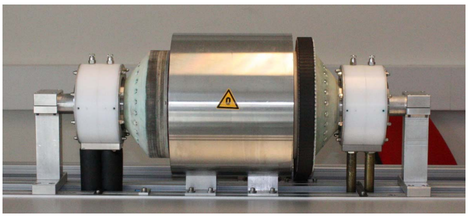
DTU Energy Conversion, Technical University of Denmark

Best results to date

- Maximum load of **1010 W at zero temperature span**
- Temperature span of **25.4 K at no-load**
- Maximum frequency **10 Hz**

World record

- Temperature span of **20.5 K at 100 W**
- Temperature span of **18.9 K at 200 W**
- Temperature span of **13.8 K at 400 W**



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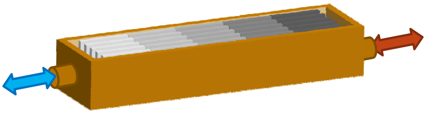
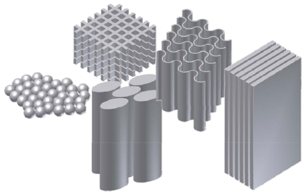
Outlook for improvements

- Careful grading of regenerator
  - Use ceramic materials
- Optimise the magnet design
- Optimise regenerator geometry
  - Ongoing postdoc project on this
- Use first order materials
  - Solve problem of hysteresis in the best materials.





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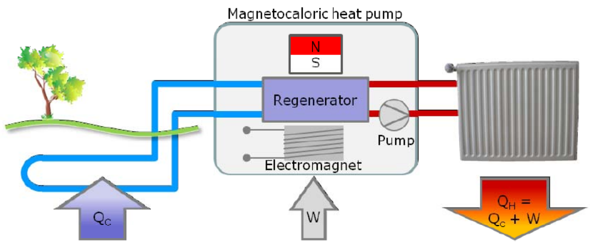
Magnetocaloric heat pumps

- A magnetocaloric device would be well suited as a heat pump – just connect the cold end with ambient, and the hot end will heat.
- The requirements on temperature span are not so high.
- Any excess heating due to friction, magnetically induced power etc. is not detrimental, but may be utilised.



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The DTU magnetic refrigeration work group

DTU Energy Conversion, Technical University of Denmark

Go back to the table of contents ⏶ or to the timetable ▲

3.2 Aquifer Thermal Energy Storage

Mikael Bastholm *(mib@ramboll.dk)*
Rambøll

Timetable ▲
Table of contents ⊼

AGENDA

- Technology (the basics)
- Above ground
- Below ground
- Integration in the collective energy supply

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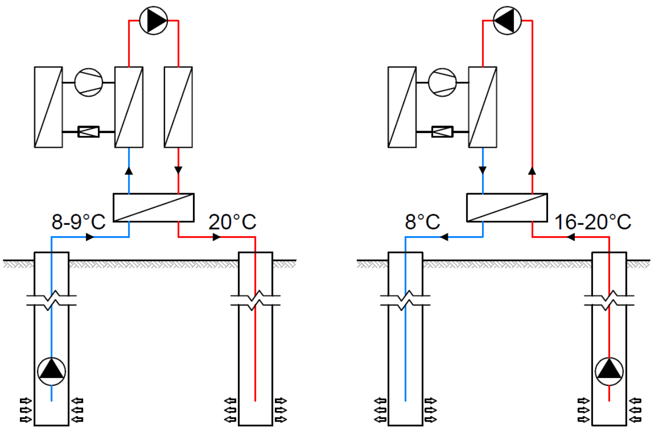
2

TECHNOLOGY (THE BASICS)



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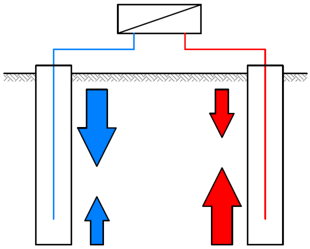
3

ENERGY BALANCE

Office building 700 kW cooling
700 kW heating



The balance:

25% kWh cooling

75% kWh heating (heat pump)

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4

QUALIFIERS FOR AQUIFER THERMAL ENERGY STORAGE

- Groundwater conditions
- Other interests in the surroundings
- Legislation/permits
- The wells
- Cooling/heating system
- The economy

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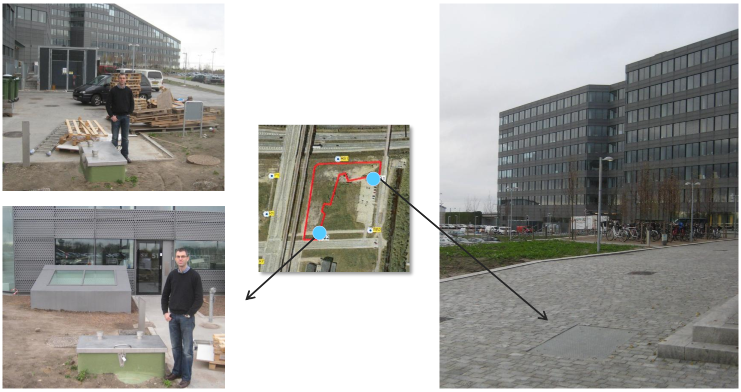
5

WHAT'S ON THE SURFACE



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6

ATES CENTRAL AT RHO – FACTS

- Comfort cooling, 2.2 MW in two machines, R717 (ammonia)
- Data warehouse cooling, 360 kW delivered by a machine, R717
- The ATES supplies chilled water in the summer and heating for the heat pumps in the winter
- Data warehouse cooling runs in the winter and also delivers heat to the heat pumps
- In case of a mishap, the data warehouse cooling system takes priority over the comfort cooling system

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7

FACTS ABOUT THE ATES SYSTEM

- Maximum cooling capacity: 1 MW of ATES systems
- Optional cooling through dry coolers in the winter
- Permission to pump:
  - Max. 90 m³/h and 500,000 m³/year
  - Temperature of operation of 9-20 degrees Celsius
  - Balance of year should be zero. The operation therefore depends much on the extent of cooling in the winter and heating in the summer
  - Effects are therefore kWh, not kW

Result at RHO :

- 370,000 kWh power saved a year
- 425 tonnes CO_2 saved annually

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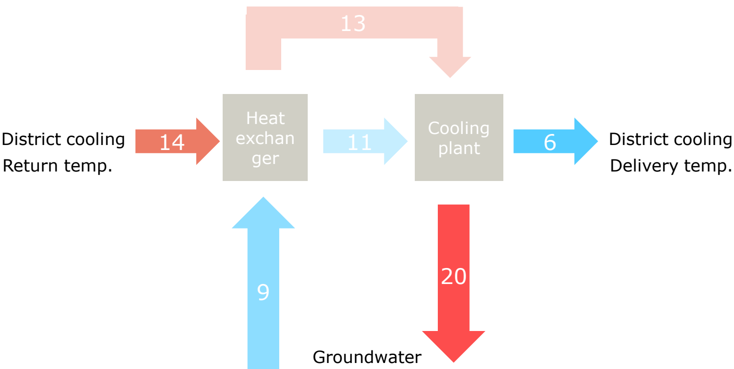
INDIRECT GROUNDWATER COOLING

- Indirect use of groundwater is also of great value



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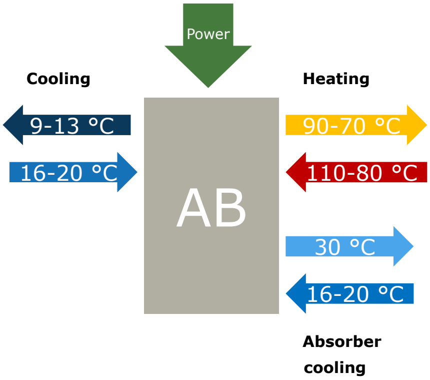
ABSORPTION



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COP VALUES

Technology	Time of operation	Power input	COP Mechanical	COP Electrical	Dry cooler Factor	Combined COP
Compressor	Out of the cooling season	1	6	6,0	1,2	5,1
	Yearly middle	1	5,5	5,5	1,2	4,7
	Peak load	1	5	5,0	1,2	4,3
Absorption with ground-water	All	0,08	0,7	8,8	-	5,7
Groundwater	All	1	50	50,0	1,0	15,3
Dry cooling	Winter	-	-	-	1,0	40,0

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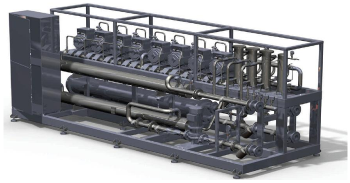
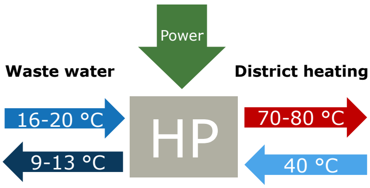
LARGE SCALE HEAT PUMPS





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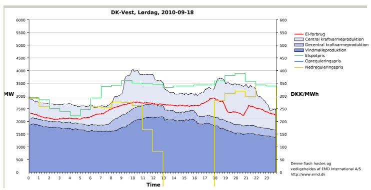
HEAT PUMPS - FLEXIBILITY

- Flexibility in electricity consumption requires storage of heat
- None or very limited options in household units
  - Would require a very large tank in the building
- Great opportunities in district energy systems
- Provides the option of integrating more wind power



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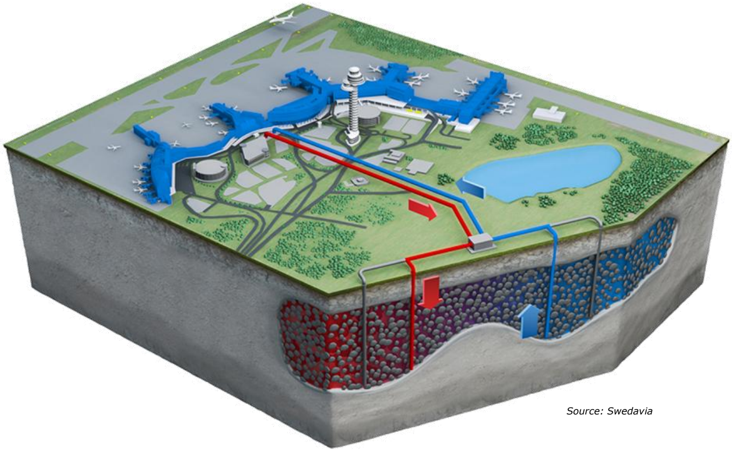
SUMMER MODE



Source: Swedavia

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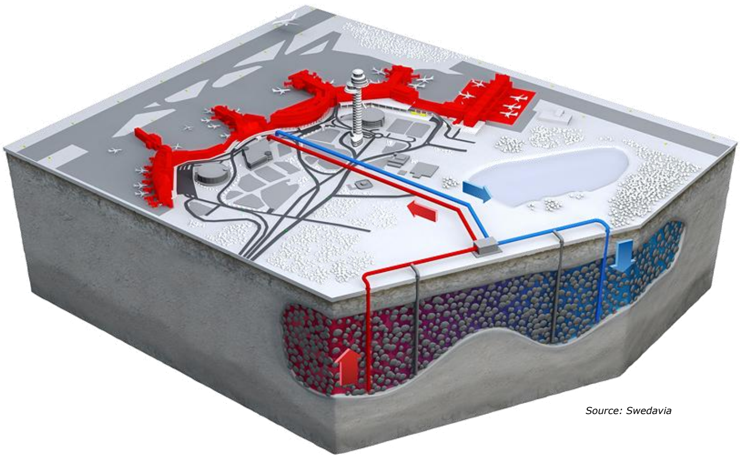
WINTER MODE



Source: Swedavia

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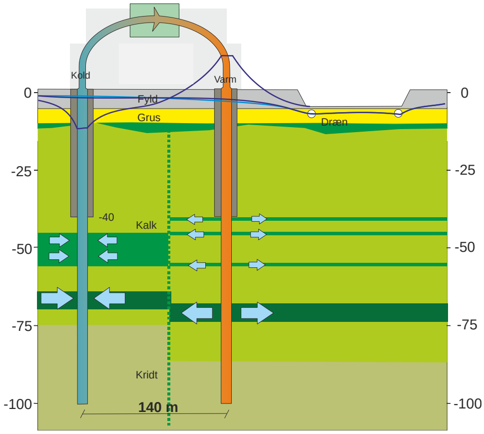
AQUIFER THERMAL ENERGY STORAGE



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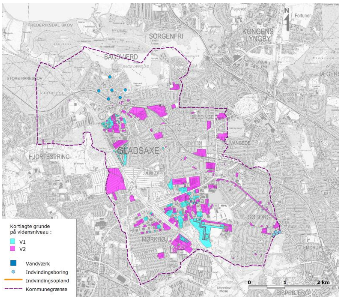
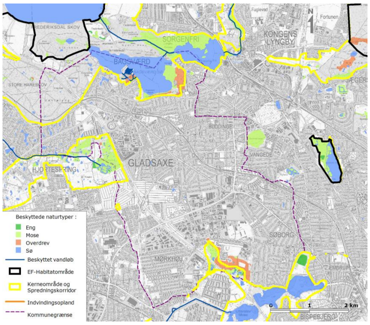
PHYSICAL RESTRICTIONS

- Contaminated soil?
- Rainwater drainage areas?





- Nature conditions?
- Protected areas?

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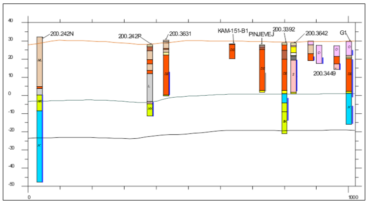
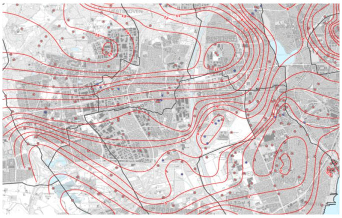
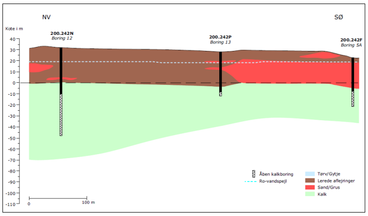
HYDROLOGICAL RESTRICTIONS

- Water flow capacity?
- Depth of groundwater levels?
- The pressure relation?
- Groundwater quality?







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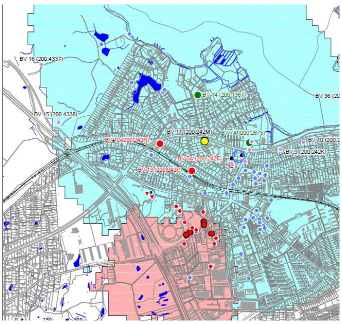
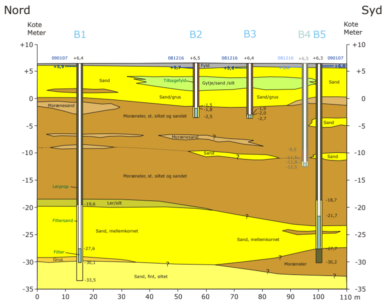
GEOTHERMAL RESTRICTIONS

Setting up and running the groundwater model





Execution of thermal modelling for extraction/injection

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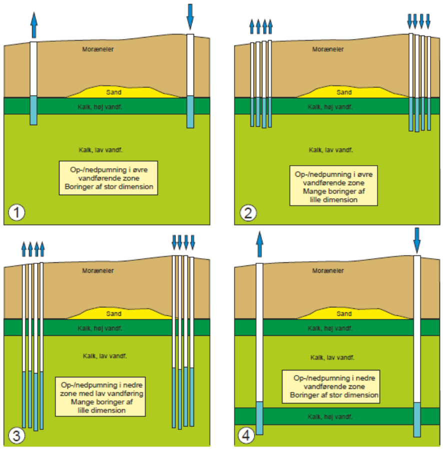
EXAMPLES OF WELL CONFIGURATIONS IN AN ATES SYSTEM



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LEGISLATION IN DENMARK

- Water Act (groundwater abstraction permit)
- Environmental Protection Act (groundwater injection permit)
- Ministry of the Environment’s "Order on heat recovery plant and groundwater cooling system"

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ATES'S INTEGRATION INTO THE PUBLIC ENERGY SUPPLY

Benefits

Increased CO2 reduction potential

Operational and socio-economic benefits

Better use of available ATES resources

Faster deployment (?)

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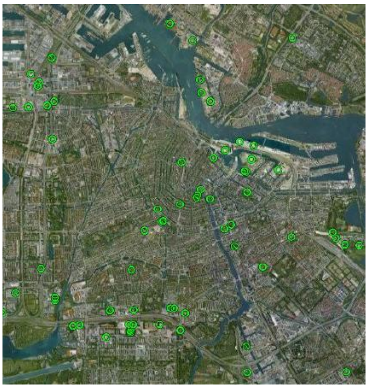
STATUS OF ATES IN DENMARK

- Individual plants
- Established after the "first come first served" principle
- Disadvantages:
  - No overarching planning
  - Poor resource utilization
  - Higher operating costs
  - Often weak operating organisations



Calje, 2010

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WHO CAN LIFT ATES INTO THE PUBLIC ENERGY SECTOR?

- Water suppliers?
- Power suppliers?

The optimal organisation:

- Water suppliers to be responsible for the operation of ATES source site
- Energy suppliers to be responsible for the optimal technical integration of ATES

Multi-suppliers!

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BARRIERS

- Water suppliers or heat suppliers:
  - Not allowed as a parallel business
- District cooling suppliers
  - Can include ATES

Is ATES a new type of utility supply?

ATES is an "energy storage technology", which sometimes delivers heating and at other times cooling

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25

NORTH HARBOUR - A CONCEPT STUDY FOR ”BY OG HAVN”

Primary objective

- Cooling in the form of district cooling

Secondary objective

- Cost and energy efficient integration of supply elements
- ATES as the "elastic band" between the district cooling and the district heating systems
- Buildings as "solar thermal collectors‟
- The recycling of heat from the generation of cooling
- Recovery of stored heat for district heating

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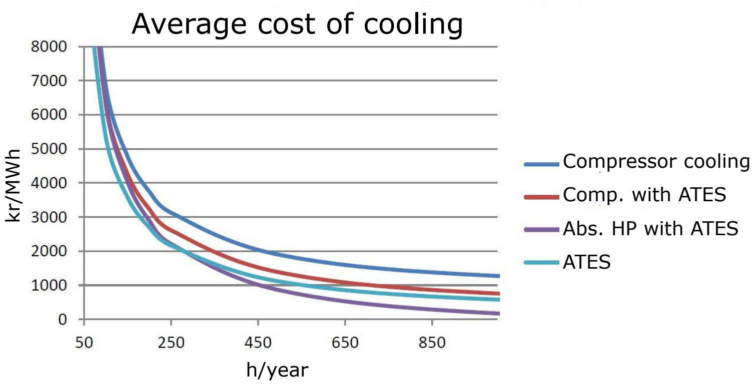
NORTH HARBOUR - A CONCEPT STUDY FOR ”BY OG HAVN”



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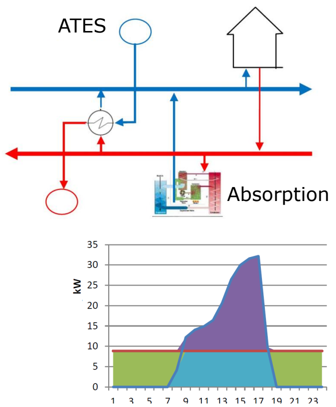
NORTH HARBOUR - A CONCEPT STUDY FOR "BY OG HAVN"

Study recommendations:

- Cooling storage for peak load
- Absorption cooling in combination with low-temperature district heating for base load
- ATES as an in-between load
- Take advantage of synergies between absorption cooling and groundwater cooling

P.S. No groundwater interests in the region



Day view of offish building

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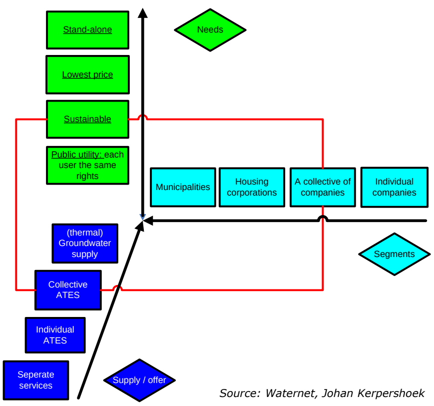
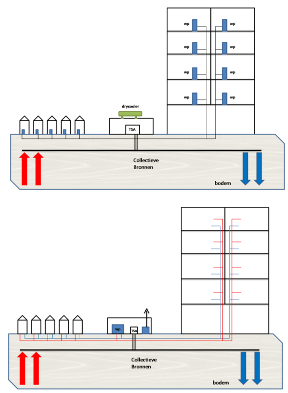
CASE – AMSTERDAM



Source: Waternet, Johan Kerpershoek



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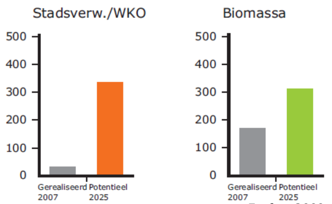
CASE – AMSTERDAM

- Target of reducing CO_2 emissions by 45% in 2025 (1990)
- The city has established a collective ATES company for climate funds
- Waternet (multi-supply) has operational responsibility
- ATES potential for displacement of 300,000 tonnes of CO_2/year in synergy with district heating



Ecofys, 2009

Back from the future
2025 - Amsterdam residents laugh at the exorbitant prices being charged for fossil fuels. Smart housing improvements have ensured that they are much less dependent on energy from the grid. Solar, wind, biomass and thermal storage provide most of their energy needs. Due to innovations in energy, they live comfortably without sky-high energy bills.

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ENVIRONMENTAL - CHALLENGES IN URBAN AREAS

Problem:

- Many groundwater contaminants
- Hydro-geological challenges
  - Water penetration in basements
  - Displacement of basement structures
- Influence of the water table in the city's lakes and streams
- Thermal effects on the groundwater

Solution:

- Carbon filter purification:
  - "Free" resource restoration
- Field studies and detailed hydraulic models
- Mutually balanced wells

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RECOMMENDATIONS

- Development and testing of energy supply concepts with ATES
- Need for a nation-wide survey of ATES
- "First come first served" principle is abandoned
- ATES in collective energy
- ATES integrated into municipalities:
  - Heat supply plans
  - Water supply plans
- The regulation of the utilities who may establish ATES should be relaxed
- Changing the tax on electricity for heat pumps in the excess production periods
- Reduction in the distribution temperature of district heating systems in areas with a potential for ATES

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PERSPECTIVES

- Once the CO_2 reduction potential and the economic benefits are known and some of the barriers overcome, ATES has the potential of spreading fast!

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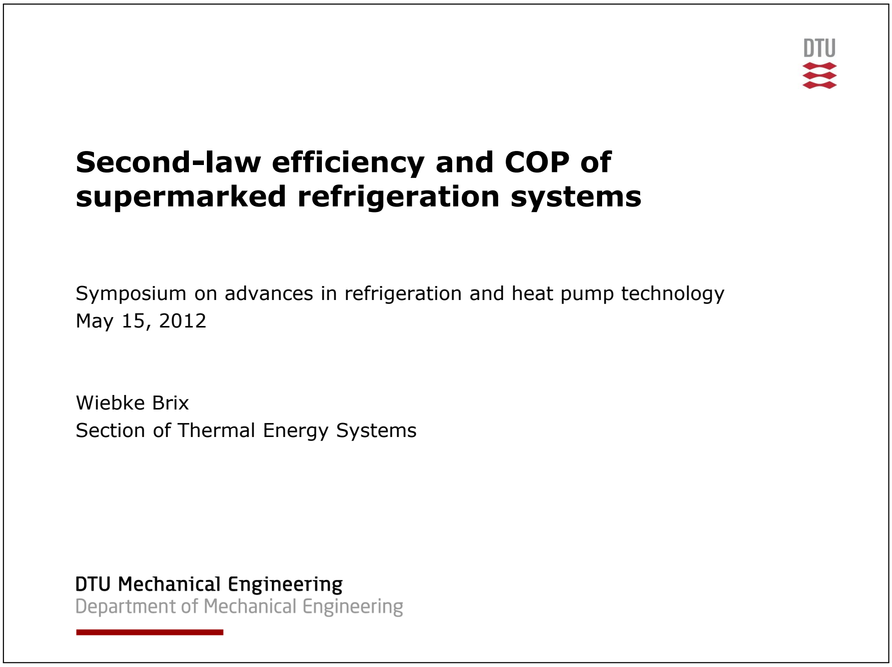
3.3 Second-law Efficiency and COP of Supermarked Refrigeration Systems

Wiebke Brix *(wb@mek.dtu.dk)*
DTU Mechanical Engineering

Timetable ▲
Table of contents ⊼



Agenda

- The problem of COP definition for systems with 3 temperature levels
- Definition of the 2nd-law efficiency
- The link between COP and 2nd-law efficiency
- Introducing a COP based on weighted heat input
- Comparison of COPs and 2nd-law efficiency for different cycles
- Comparison of different refrigerants
- Conclusions

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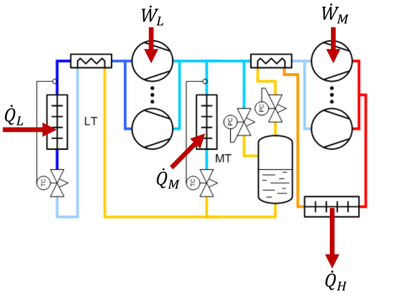
The problem of COP definition

Loads are at different temperature levels.



$$\text{COP} = \frac{\text{What we get}}{\text{What we pay}}$$

?

$$COP = \frac{\dot{Q}_L + \dot{Q}_M}{\dot{W}_L + \dot{W}_M}$$

- Is this the 'correct' way of defining performance for systems with loads at different temperatures?
- Would it be more 'correct' to take the different temperature levels into account?

PI-diagrams from PackCalc software, www.ipu.dk

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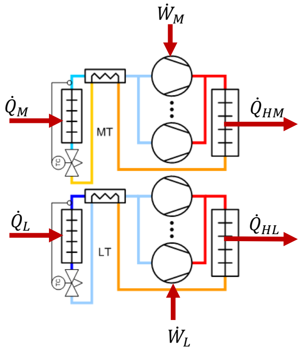
COP of two one-stage systems



Weighted by load: $COP = \left(\frac{\dot{Q}_L}{\dot{Q}_L+\dot{Q}_M}\right)\cdot COP_{LT} + \left(\frac{\dot{Q}_M}{\dot{Q}_L+\dot{Q}_M}\right)\cdot COP_{MT}$

Weighted by power input: $COP = \left(\frac{\dot{W}_L}{\dot{W}_L+\dot{W}_M}\right)\cdot COP_{LT} + \left(\frac{\dot{W}_M}{\dot{W}_L+\dot{W}_M}\right)\cdot COP_{MT}$

$$= \frac{\dot{Q}_L + \dot{Q}_M}{\dot{W}_L + \dot{W}_M}$$

Or something different?

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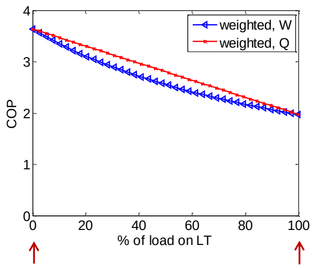
Example: COP of two one-stage systems

- R290 in both cycles

Conditions:

Room temperature at LT	-20° C
Room temperature at MT	2° C
Ambient temperature	30° C
Isentropic efficiencies	70%
Superheat	10 K
Subcooling	2 K
Pinch in evaporators	5 K
Pinch in condenser	7 K



MT system only LT system only

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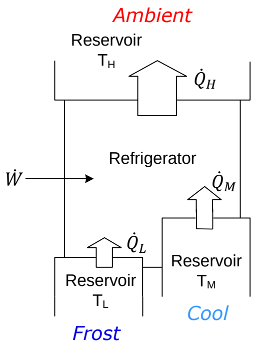
2nd-law efficiency (1)



1st: $\dot{Q}_L + \dot{Q}_M - \dot{Q}_H + \dot{W} = 0$

2nd: $\dot{S}_{gen} = \frac{\dot{Q}_H}{T_H} - \frac{\dot{Q}_L}{T_L} - \frac{\dot{Q}_M}{T_M} \geq 0$

Eliminating $\dot{Q}_H$ and combining the two equations:

$$\dot{W} = \dot{Q}_L\left(\frac{T_H}{T_L} - 1\right) + \dot{Q}_M\left(\frac{T_H}{T_M} - 1\right) + \dot{S}_{gen} T_H$$

Reversible operation:

$$\dot{W}_{rev} = \dot{Q}_L\left(\frac{T_H}{T_L} - 1\right) + \dot{Q}_M\left(\frac{T_H}{T_M} - 1\right)$$

Extra power due to irreversibility:

$$\dot{W}_{lost} = \dot{W} - \dot{W}_{rev} = T_H \dot{S}_{gen} \geq 0$$

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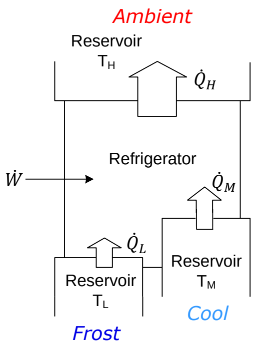
2nd-law efficiency (2)



Exergy:

Exergy content of supplied power: $\dot{E}_W = \dot{W}$

Exergy content of heat transfer:

$$\dot{E}_{Q_L} + \dot{E}_{Q_M} = \dot{Q}_L\left(\frac{T_H}{T_L} - 1\right) + \dot{Q}_M\left(\frac{T_H}{T_M} - 1\right) \quad = \dot{E}_{W,rev}$$

2nd-law efficiency:

$$\eta_{II} = \frac{-\dot{E}_{W,rev}}{-\dot{E}_W} = \frac{-\left(\dot{E}_{Q_L} + \dot{E}_{Q_M}\right)}{-\dot{E}_W}$$

$$= \frac{\dot{Q}_L\left(\frac{T_H}{T_L} - 1\right) + \dot{Q}_M\left(\frac{T_H}{T_M} - 1\right)}{\dot{W}}$$

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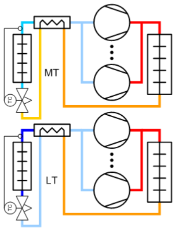
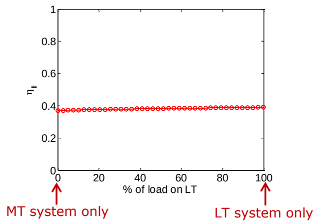
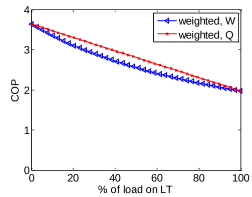
Example: 2nd-law efficiency

- R290 in both cycles







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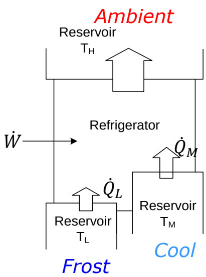
Link between COP and 2nd-law efficiency

- For a single-stage system: $\eta_{II} = \dfrac{COP}{COP_C}$ where $COP_C = \dfrac{T_L}{T_H - T_L}$

- What is the Carnot COP of a system with temperatures at different loads?



Reversible machine: $$COP_C = \frac{\dot{Q}_L + \dot{Q}_M}{\dot{Q}_L\left(\frac{T_H}{T_L} - 1\right) + \dot{Q}_M\left(\frac{T_H}{T_M} - 1\right)}$$

$$\rightarrow \quad COP = COP_C \cdot \eta_{II} = \frac{\dot{Q}_L + \dot{Q}_M}{\dot{W}_L + \dot{W}_M}$$

But there is still the problem of adding heat at different temperature levels!

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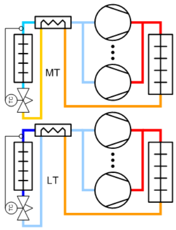
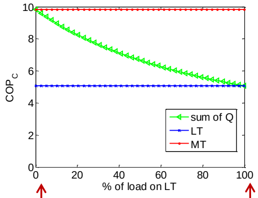
Example: Carnot COP of two 1-stage cycles

- R290 in both cycles





MT system only LT system only

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A different approach...

- Temperature difference between L and H sets the limits for the system

$$\rightarrow COP_{C,L} = \frac{T_L}{T_H - T_L}$$ is used as reference

$$\rightarrow COP_{II} = \eta_{II} \cdot COP_{C,L} = \frac{\dot{Q}_L + \dot{Q}_M \dfrac{COP_{C,L}}{COP_{C,M}}}{\dot{W}_L + \dot{W}_M}$$

Heat input is weighted

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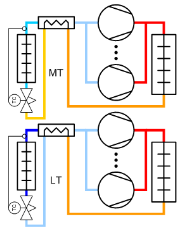
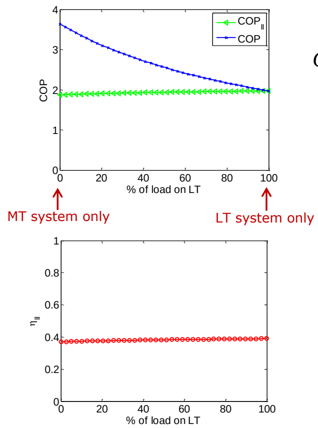
Example: COP_{II} of two 1-stage cycles

- R290 in both cycles





$$COP_{II} = \frac{\dot{Q}_L + \dot{Q}_M \left(\frac{COP_{C,L}}{COP_{C,M}} \right)}{\dot{W}_L + \dot{W}_M}$$

$$COP = \frac{\dot{Q}_L + \dot{Q}_M}{\dot{W}_L + \dot{W}_M}$$

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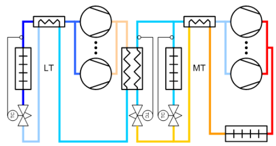
Comparison of different two-stage systems

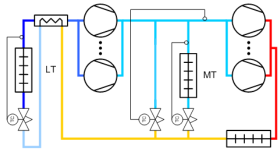
Conditions:

Room temperature at LT	-20° C
Room temperature at MT	2° C
Ambient temperature	30° C
Isentropic efficiencies	70%
Superheat	10 K
Subcooling	2 K
Pinch in evaporators	5 K
Pinch in condenser	7 K
Pinch in cascade HEX	3 K

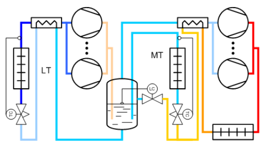
Cascade



Liquid injection



Open intercooler



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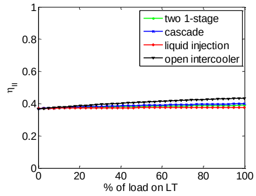
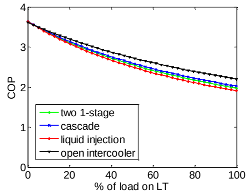
Comparison of different two-stage systems

- R290 in all cycles

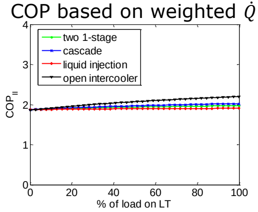


2nd-law efficiency

'traditional' COP



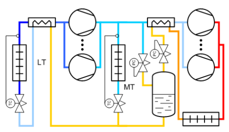
COP based on weighted $\dot{Q}$



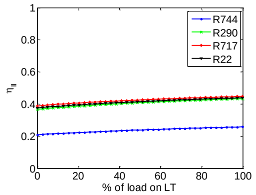
Comparison of different refrigerants

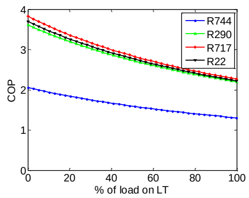
- System: Open intercooler 2-stage

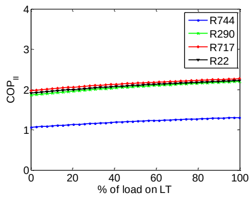
except R744:



(gas cooler pressure optimized, flash tank at 15° C)







Conclusion

- The 2nd-law efficiency is a useful figure of merit for a 3 temperature level system.
- There is a link between 2nd-law efficiency and COP. Definition of the Carnot COP is central here.
- Using the largest temperature difference as reference results in a COP where the heat input is weighted.
- Comparison of different systems and different refrigerants can be made equally well by the 'traditional' COP, the COP based on weighted heat input and the 2nd-law efficiency.

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Second-law efficiency and COP of supermarked refrigeration systems

Symposium on advances in refrigeration and heat pump technology
May 15, 2012

Wiebke Brix
Section of Thermal Energy Systems

DTU Mechanical Engineering
Department of Mechanical Engineering

Go back to the table of contents ⯒ or to the timetable ▲

3.4 Optimization of Commercial Refrigeration Plants

Christian Heerup *(chp@teknologisk.dk)*
DTI, Energy & Climate

Timetable ▲
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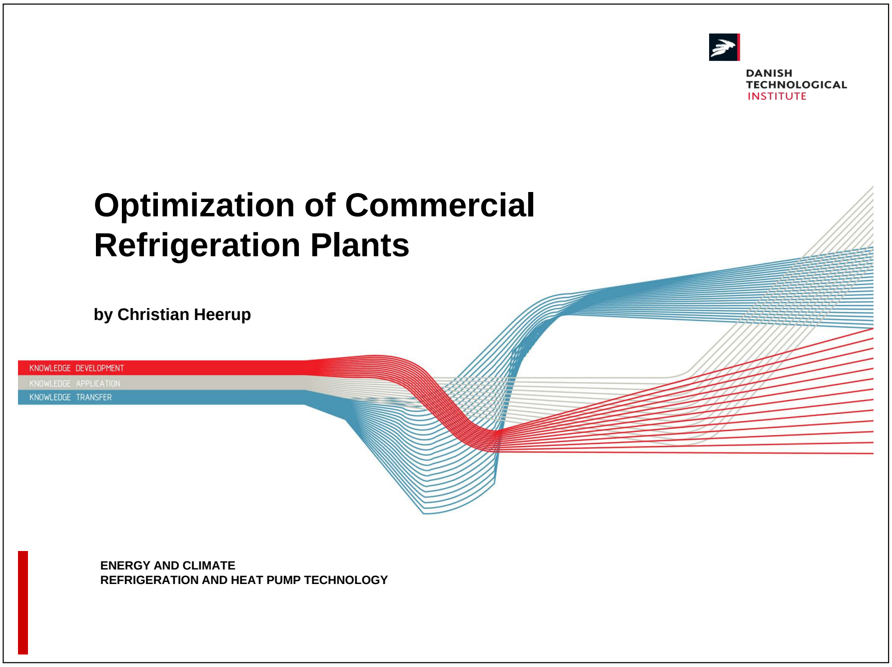


ESO2 Optimization of super market refrigeration systems

DANISH TECHNOLOGICAL INSTITUTE

Main objectives

- Software tool for sizing compressors and estimating load profile
- Establishing actual refrigeration capacity needed and identifying saving potential
- Software for diagnosing energy performance of plant
- Optimized energy performance by coordinating/ overriding local controllers as an add on for the present control system

Status for activities

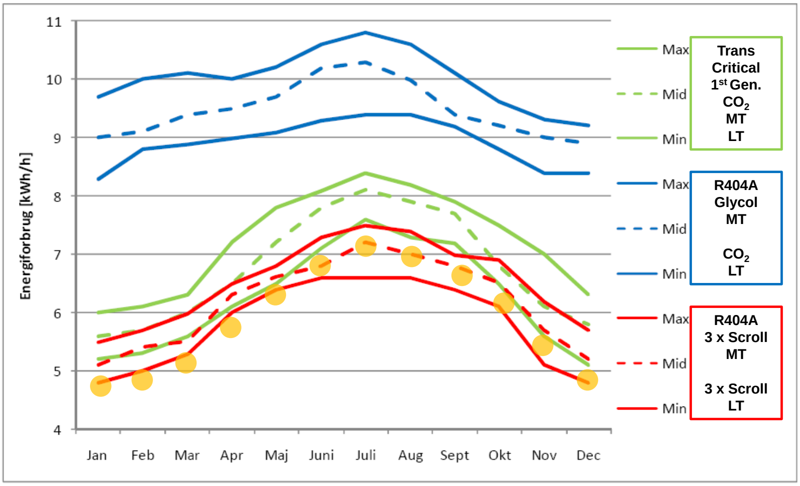
- Model using exergy calculations for identifying mal-performance
- Documentation of performance with standard settings (data logging)
- Analysis of dynamics/ data processing

ENERGY AND CLIMATE
REFRIGERATION AND HEAT PUMP TECHNOLOGY

2

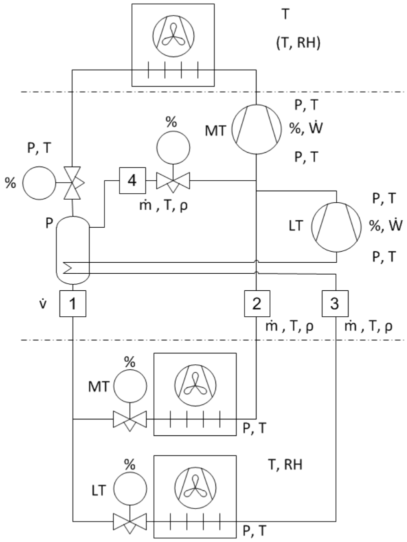
Fakta Reference Installations

DANISH TECHNOLOGICAL INSTITUTE



REFRIGERATION AND HEAT PUMP TECHNOLOGY

3



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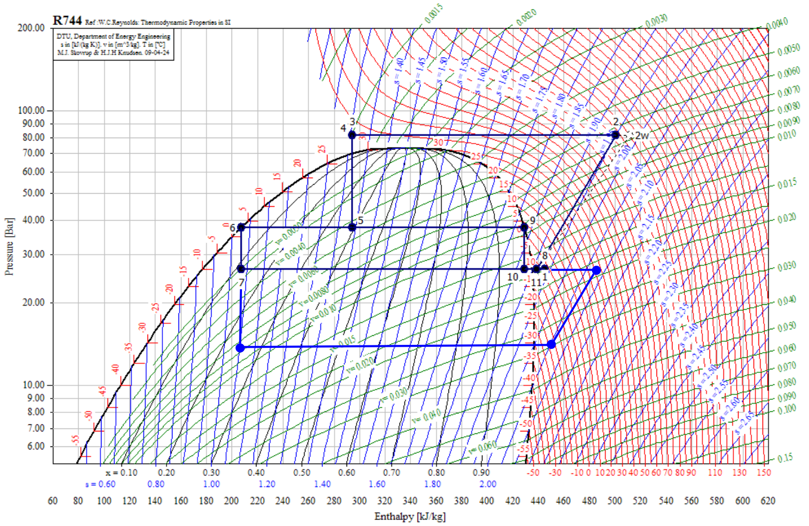
CO_2 Booster Pack

- MT 30 kW
  - 7 evaporators
  - 2 compressors (1 inverter)
- LT 8 kW
  - 4 evaporators
  - 2 compressors

ENERGY AND CLIMATE
REFRIGERATION AND HEAT PUMP TECHNOLOGY

4



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5

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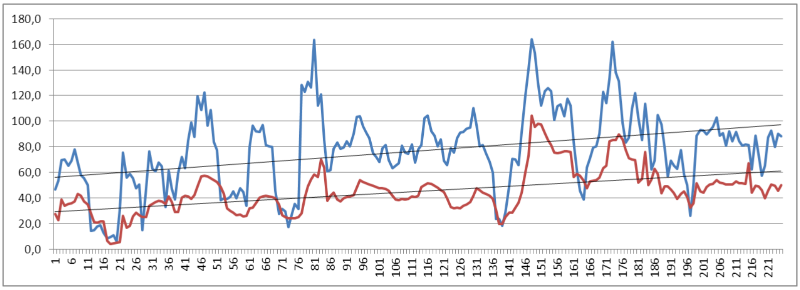
Coriolis versus Vortex flow sensor



ENERGY AND CLIMATE
REFRIGERATION AND HEAT PUMP TECHNOLOGY

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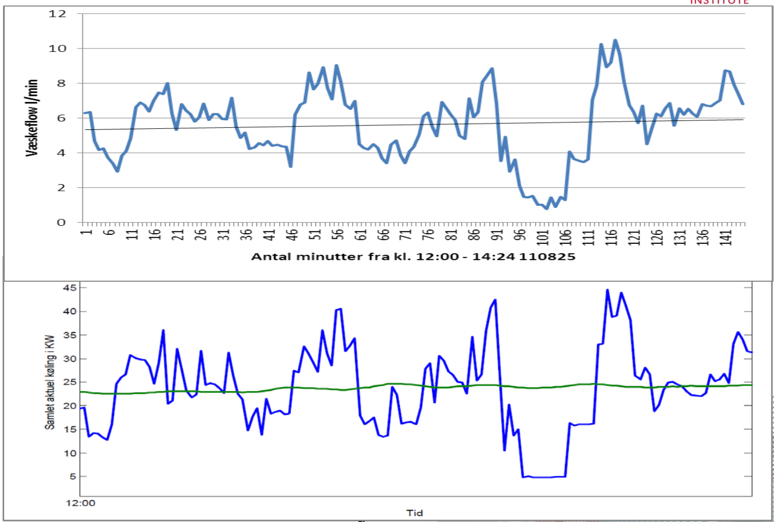
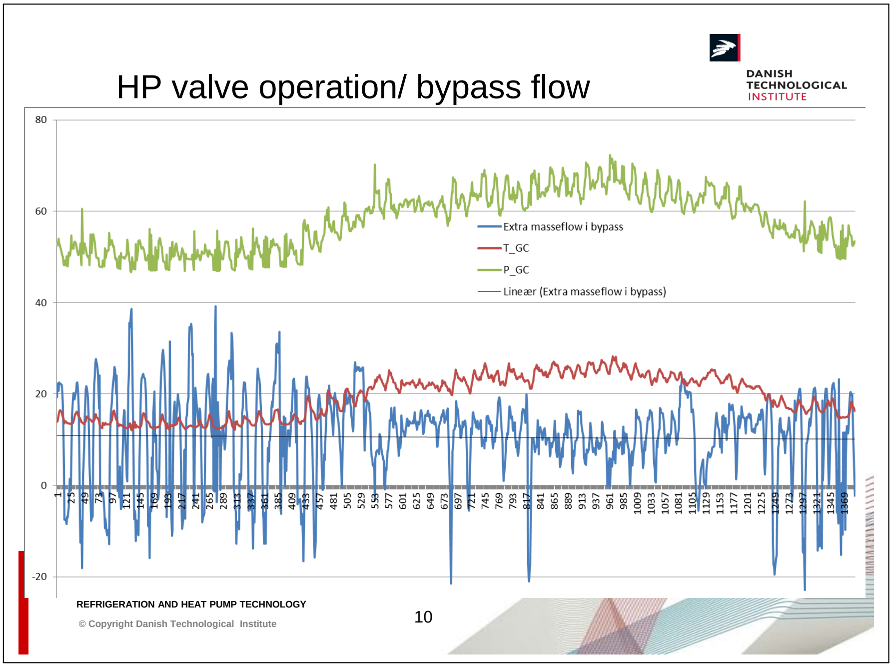
Valve capacity versus Vortex flow sensor




HP valve operation/ bypass flow
DANISH TECHNOLOGICAL INSTITUTE
Extra masseflow i bypass
T_GC
P_GC
Lineær (Extra masseflow i bypass)
REFRIGERATION AND HEAT PUMP TECHNOLOGY
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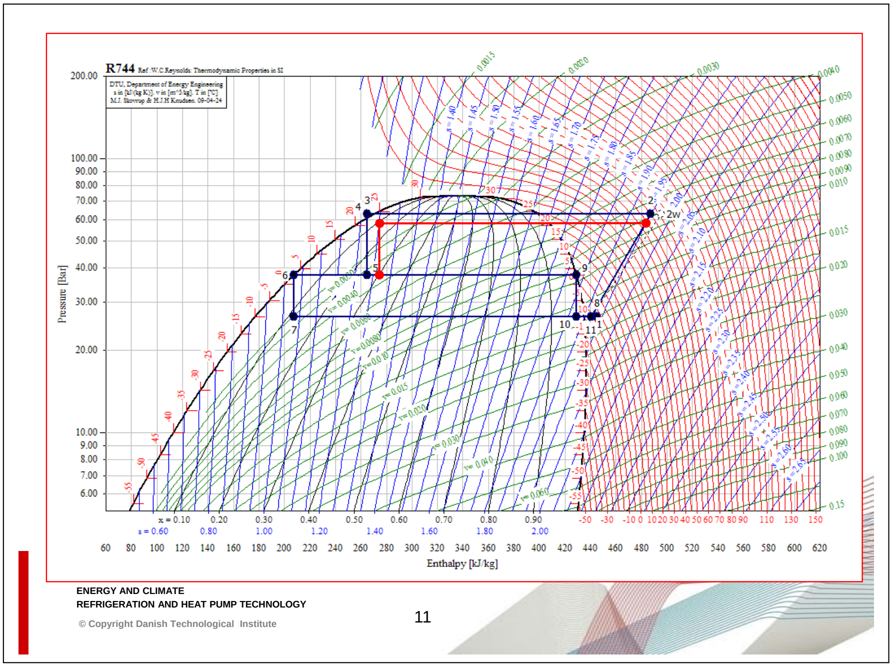
R744
Pressure [Bar]
Enthalpy [kJ/kg]
ENERGY AND CLIMATE
REFRIGERATION AND HEAT PUMP TECHNOLOGY
© Copyright Danish Technological Institute
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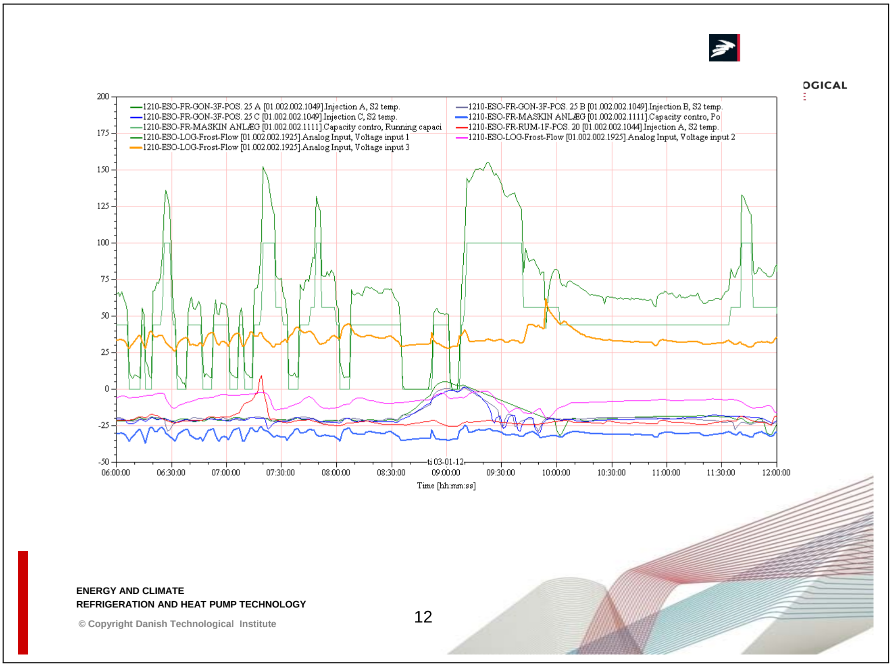

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1210-ESO-FR-GON-3F-POS. 25 C [01.002.002.1049].Injection C, S2 temp.
1210-ESO-FR-MASKIN ANLÆG [01.002.002.1111].Capacity contro, Running capaci
1210-ESO-LOG-Frost-Flow [01.002.002.1925].Analog Input, Voltage input 1
1210-ESO-LOG-Frost-Flow [01.002.002.1925].Analog Input, Voltage input 3
1210-ESO-FR-GON-3F-POS. 25 B [01.002.002.1049].Injection B, S2 temp.
1210-ESO-FR-MASKIN ANLÆG [01.002.002.1111].Capacity contro, Po
1210-ESO-FR-RUM-1F-POS. 20 [01.002.002.1044].Injection A, S2 temp.
1210-ESO-LOG-Frost-Flow [01.002.002.1925].Analog Input, Voltage input 2
200
175
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125
100
75
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25
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-25
-50
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Time [hh:mm:ss]
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REFRIGERATION AND HEAT PUMP TECHNOLOGY
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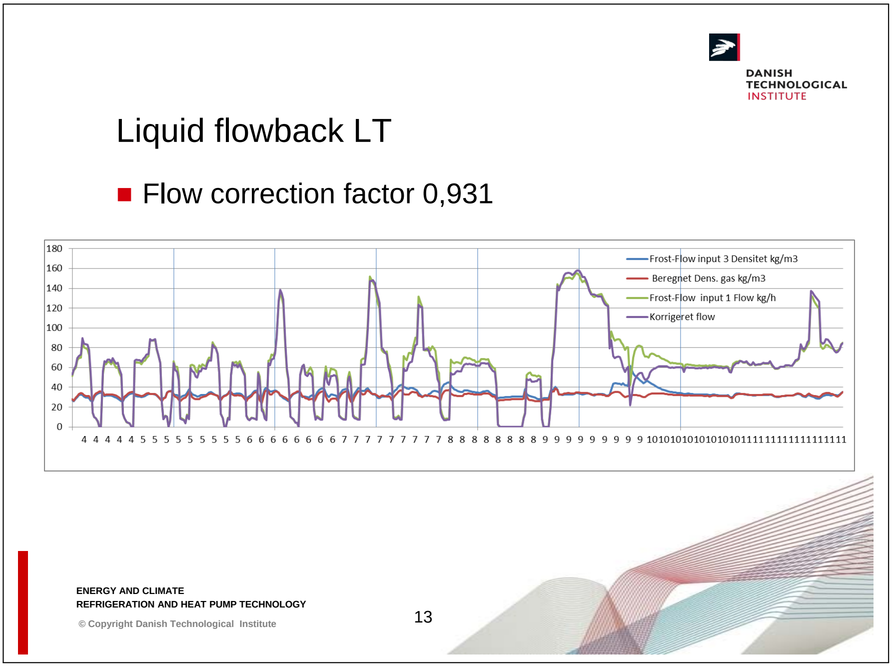

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TECHNOLOGICAL
INSTITUTE
Liquid flowback LT
Flow correction factor 0,931
Frost-Flow input 3 Densitet kg/m3
Beregnet Dens. gas kg/m3
Frost-Flow input 1 Flow kg/h
Korrigeret flow
180
160
140
120
100
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60
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20
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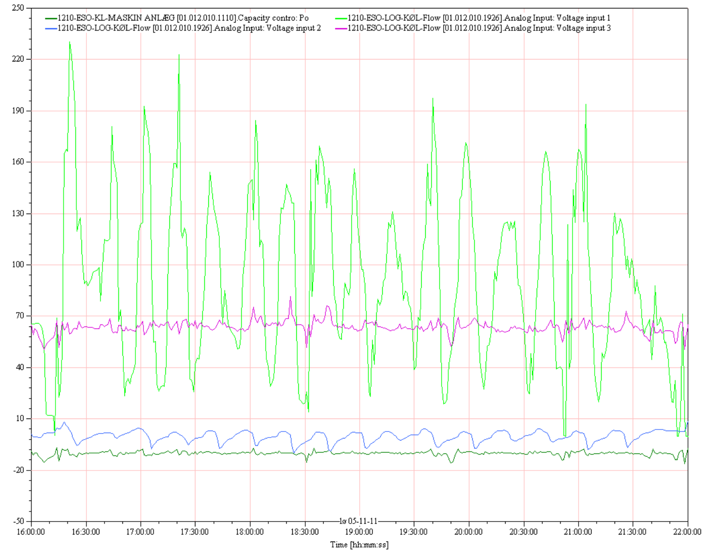

OGICAL

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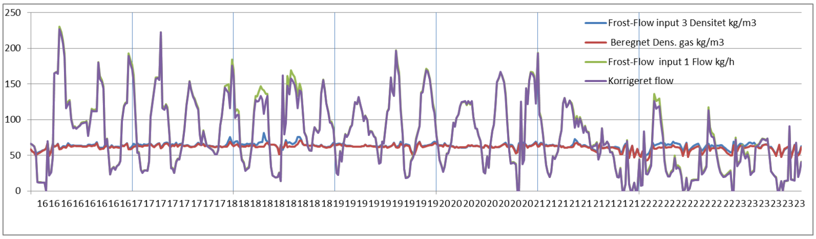
Liquid flowback MT

- Flow correction factor 0,985



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Open vertical cabinet for fresh products

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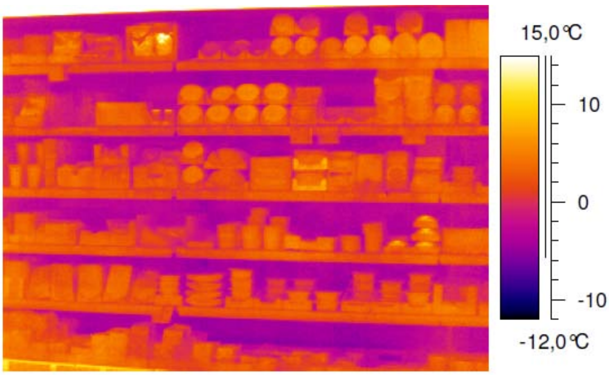
Thermografic analysis of vertical cabinet



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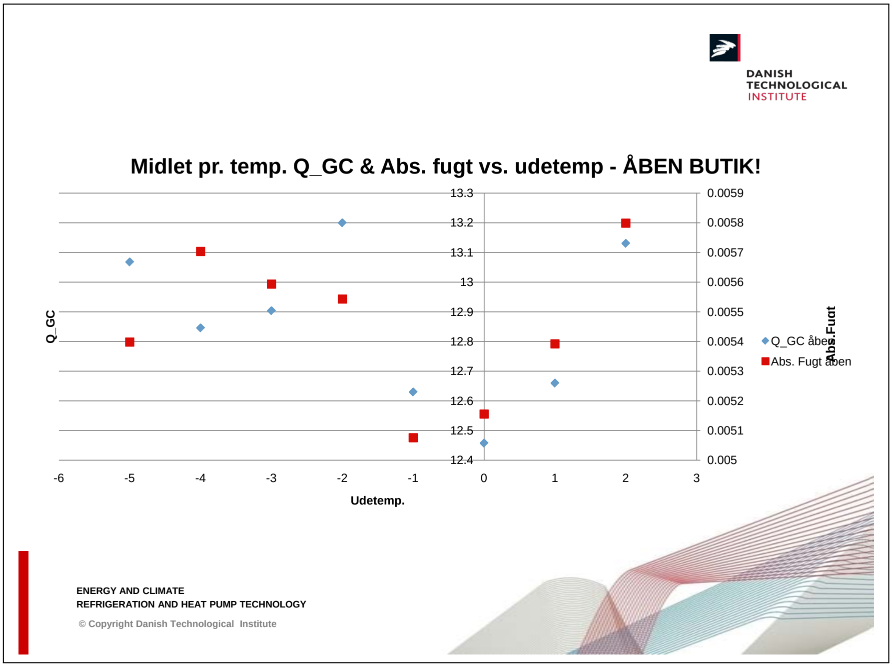
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Abs.Fugt
Udetemp.
Q_GC åben
Abs. Fugt åben
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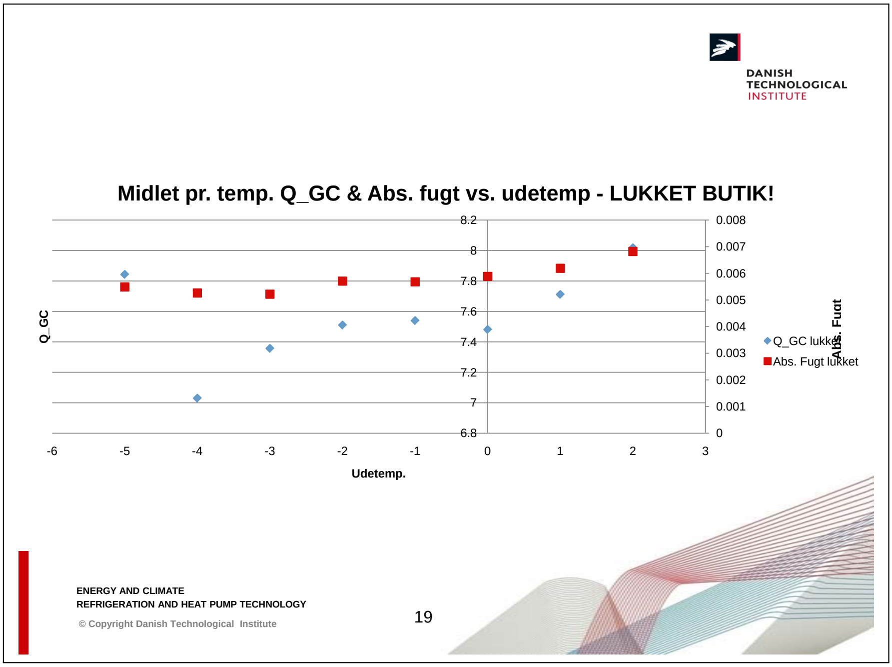
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Abs. Fugt lukket
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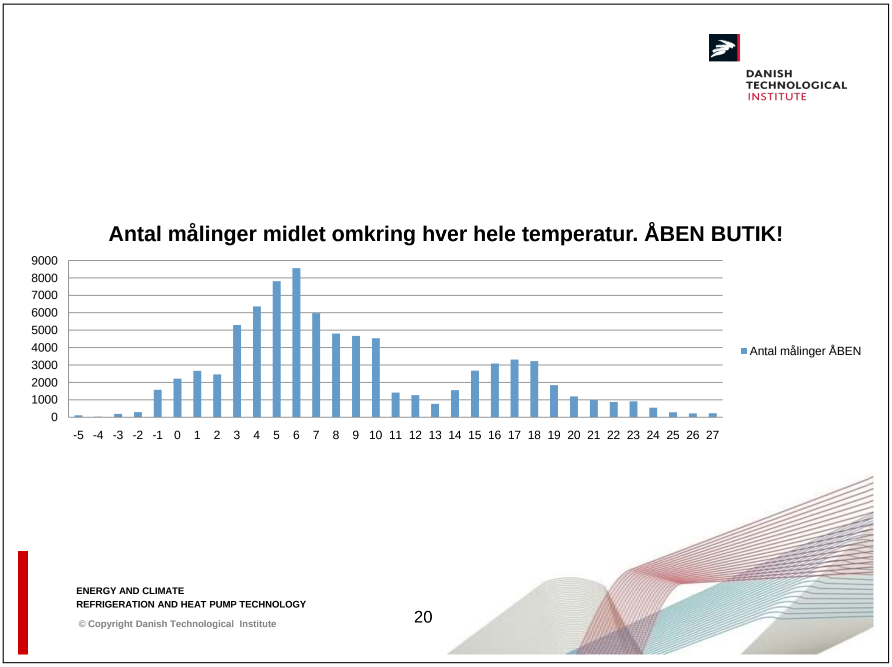
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Antal målinger midlet omkring hver hele temperatur. ÅBEN BUTIK!
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Antal målinger ÅBEN
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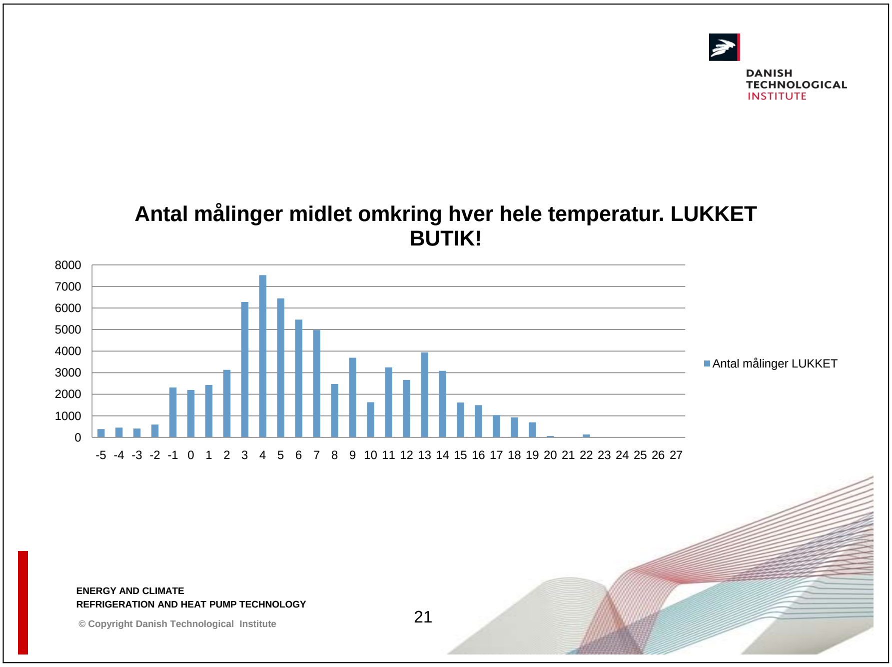
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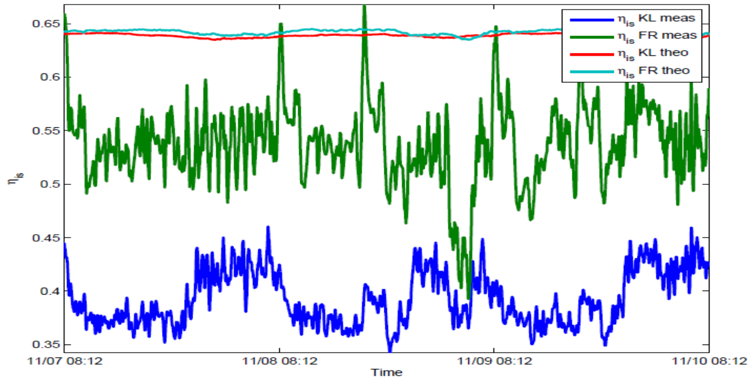
Estimation of isentropic efficiency



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Project Organisation

Funding
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Danfoss
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DTU Informatik
DTU Mekanik
IPU
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Super Køl

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3.5 Compensation of Airflow Maldistribution in Fin-and-Tube Evaporators

Martin Ryhl Kærn *(pmak@mek.dtu.dk)*
DTU Mechanical Engineering

Timetable ▲
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1. Introduction
   - Objective
   - System of interest
   - Experiments and circuitry modifications
2. Model development
3. Simulation setup
4. Results
   - Experimental validation
   - Airflow maldistribution and compensation
   - Possible area savings
5. Conclusion
6. Questions

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What is maldistribution?

- Occurs in multi-channel evaporators
- Results in uneven superheated regions and consequently:
  - Capacity reduction
  - Higher power consumption
  - Higher material usage
- Can be caused by:
  - Non-uniform airflow, temperature or humidity distribution
  - Fouling
  - Non-uniform liquid/vapor inlet distribution
  - Feeder tube bending
  - Improper heat exchanger design or installation

Objective

The objective of this study is:

- To study and validate the hypothesis from Kærn *et al.* (2011c), which was based on a numerical study:

  The face split evaporator performs better than the interlaced evaporator if flow maldistribution is compensated by control of individual channel superheat

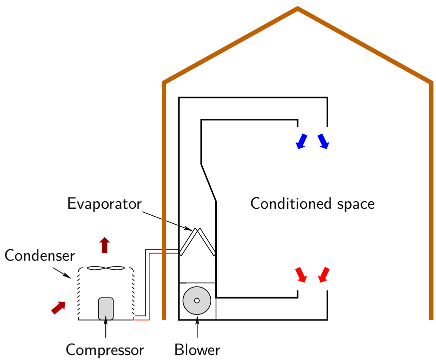
Kærn, M. R., Elmegaard, B., Larsen, L. F. S., 2011c. Comparison of fin-and-tube interlaced and face split evaporators with flow mal-distribution and compensation. *In proceedings of the 23th International Congress of Refrigeration.* IIR/IIF, Prague, Czech Republic.

Objective

To investigate this hypothesis we:

- Validate the numerical model from Kærn *et al.* (2011c) with experiments of both designs (interlaced and face split)
- Study airflow maldistribution and compensation potential
  - Using both linear velocity and CFD predicted profile
  - In both wet and dry conditions
  - And on larger evaporators (from 2 rows to 3 rows)
- And study the possible area savings experimentally
  - By cutting of face splits

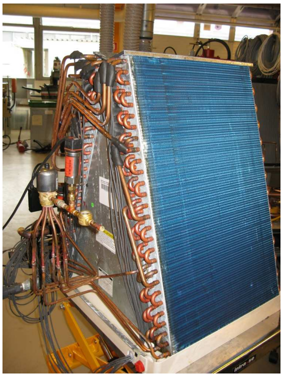
System of interest

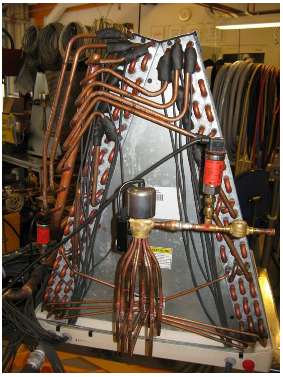


Experiments and circuitry

Experiments from Danfoss, Nordborg, Denmark using EcoFlow™

The interlaced circuitry



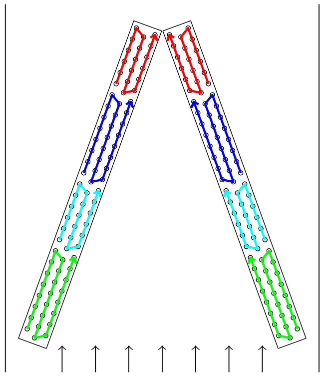


8/35 Mechanical Engineering Compensation of airflow maldistribution in F&T evaporators May 15, 2012

Experiments and circuitry

The circuitry

Face split



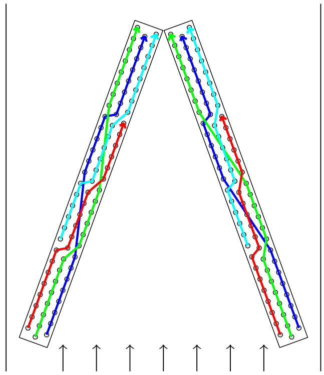
Airflow

9/35 Mechanical Engineering Compensation of airflow maldistribution in F&T evaporators May 15, 2012

Experiments and circuitry

The circuitry

Interlaced



Mostly used today

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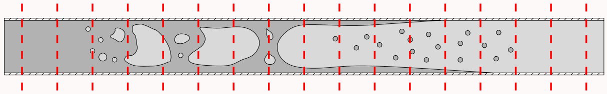
Model development

The evaporator model is a

Dynamic one-dimensional distributed mixture model



$$A\frac{\partial \bar{\rho}}{\partial t} + \frac{\partial \dot{m}}{\partial z} = 0$$

$$\frac{\partial \dot{m}}{\partial t} + \frac{\partial}{\partial z}\left(\frac{\dot{m}^2}{\rho' A}\right) = -A\frac{\partial p}{\partial z} - F_w A - \bar{\rho} g A \sin\theta$$

$$A\frac{\partial}{\partial t}\left(\overline{\bar{\rho} h} - p\right) + \frac{\partial}{\partial z}\left(\dot{m}h\right) = P q''_w$$

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Model development

Overview of used relations and correlations

Air-side	
Heat transfer	Wang et al. (1999)
Mass transfer	The Colburn analogy
Fin efficiency	Schmidt (1949), (Schmidt approximation)
Two-phase	
Heat transfer	Shah (1982)
Friction	Müller-Steinhagen and Heck (1986)
Bend friction	Geary (1975)
Single phase	
Heat transfer	Gnielinski (1976)
Friction	Blasius (2002)
Bend friction	Ito (1960)

where effectiveness-NTU relations for cross-flow HX are applied

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Model development

The main assumptions are

- Each coil is in similar flow maldistribution conditions
- Negligible tube to tube heat conduction through fins
- Air flows one-dimensionally and perpendicular through coil

Outline

Simulation setup

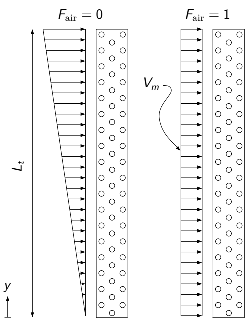
The system of interest is an 17.58 kW R410A unit
The conditions used follows from

Inlet air temperature	26.7°C
Inlet wet-bulb temperature	19.4°C
Air flow rate	0.85 m^3s^{-1} (1800 CFM)
Liquid temperature before expansion	46°C
Superheat	5 K
Compressor volume flow rate	11.3 m^3h^{-1}

Simulation setup

To simulate maldistribution a distribution parameter is defined

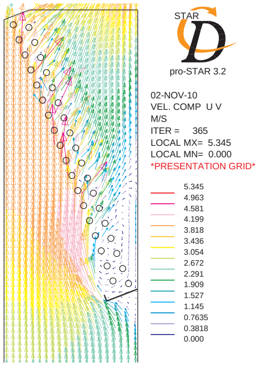
$$V(y) = V_m F_{\mathrm{air}} + \left(\frac{2V_m(1-F_{\mathrm{air}})}{L_t}\right) y$$

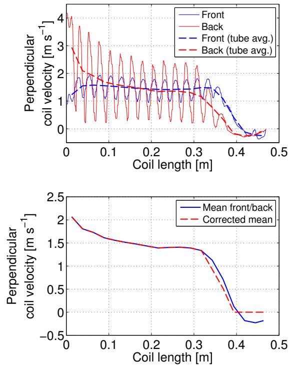


Simulation setup

Additionally, a CFD predicted velocity profile is used

- extrapolated from 8.8 kW to 17.58 kW size (Kærn, 2011d, PhD thesis)



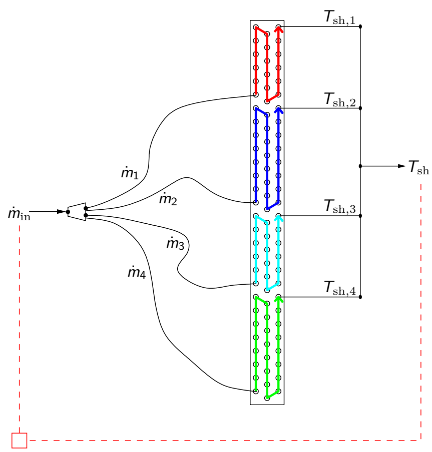


Simulation setup

Control

Without compensation

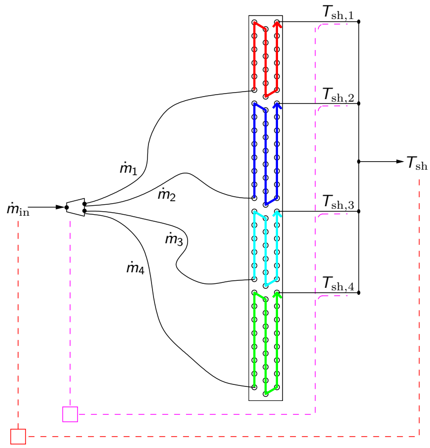
typical EXV or TXV



Simulation setup

Control

With compensation



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Experimental validation

The validation is performed in wet conditions with compensation using the CFD predicted profile

	Interlaced		Face split	
	Experiment	Model	Experiment	Model
Cooling capacity (air-side)	16.04 kW	15.46 kW	15.56 kW	15.55 kW
Cooling capacity (ref-side)	15.67 kW	15.46 kW	16.01 kW	15.55 kW
Mass flow rate	0.0931 kg/s	0.0918 kg/s	0.0955 kg/s	0.0929 kg/s
Evaporator outlet pressure	10.53 bar	10.39 bar	10.73 bar	10.46 bar
Sensible heat	12.57 kW	12.68 kW	12.27 kW	12.93 kW
Latent heat	3.41 kW	2.78 kW	3.24 kW	2.62 kW

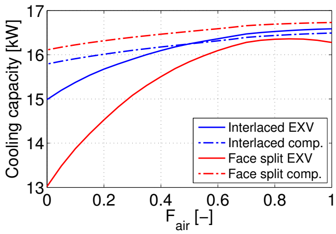
Observations:

- It is hard to justify the best circuitry at compensation, because the standard uncertainty in these experiments is ± 5%
- However the model results agree well, and we may use the model to analyze the circuitry effects in more detail

Airflow maldistribution and compensation

Dry conditions using linear profile

Capacity



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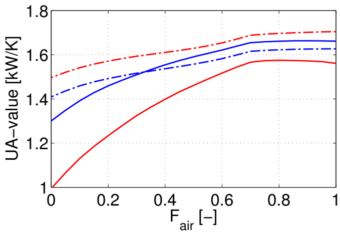
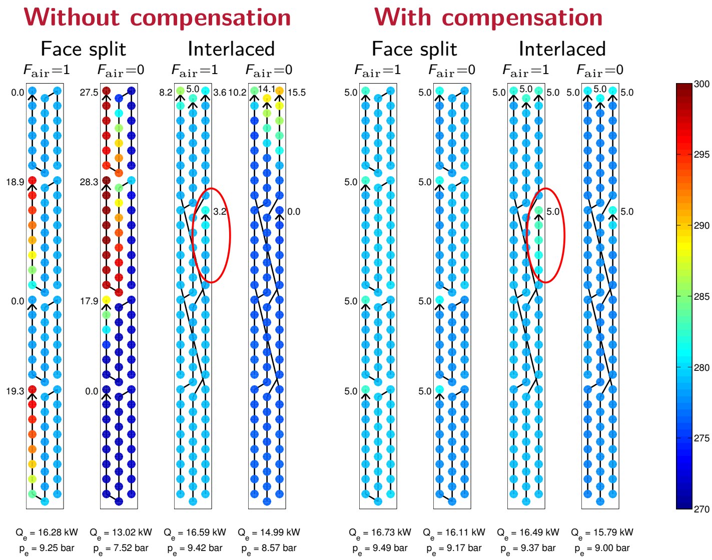


Without compensation / With compensation



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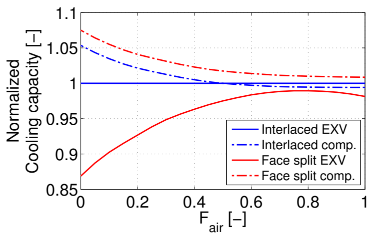
Airflow maldistribution and compensation

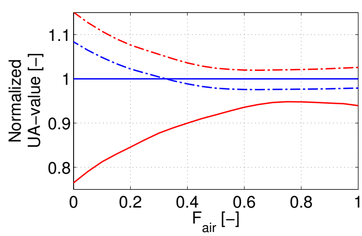
Dry conditions using linear profile

Normalized capacity



$F_{\mathrm{air}} = 0 \Rightarrow +7.5\%$

Normalized UA



$F_{\mathrm{air}} = 0 \Rightarrow +15\%$

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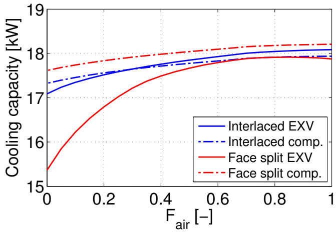
Airflow maldistribution and compensation

Wet conditions using linear profile

Capacity



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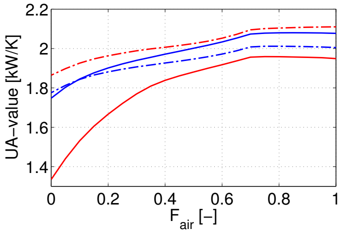

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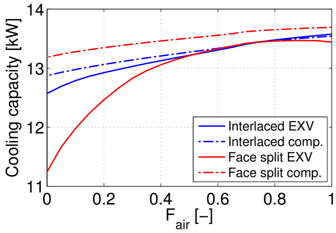
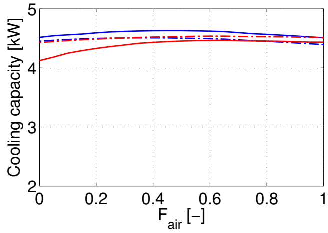
Airflow maldistribution and compensation

Wet conditions using linear profile

Sensible heat



Latent heat



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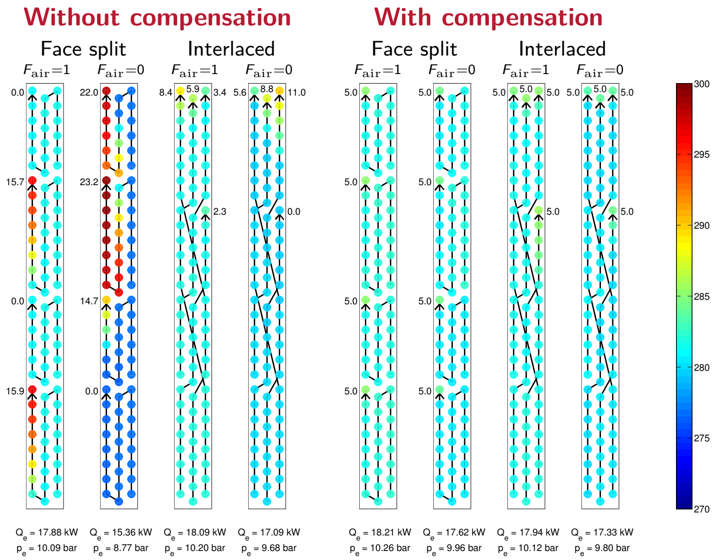

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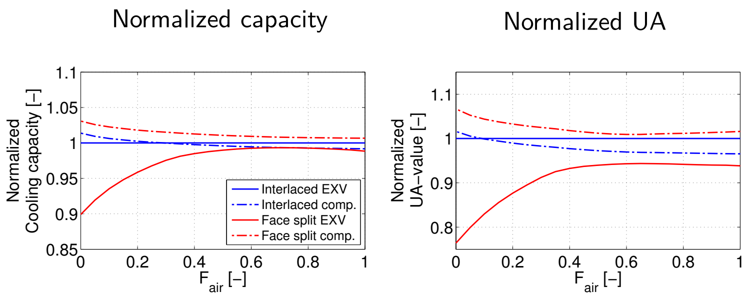
Airflow maldistribution and compensation

Wet conditions using linear profile



$F_{air} = 0 \Rightarrow +3.1\%$ $F_{air} = 0 \Rightarrow +6.7\%$

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Airflow maldistribution and compensation

CFD predicted profile

Dry conditions

	$\dot{Q}_e$ [kW]	$\dot{Q}_e$ [%-change]	UA [kW/K]	UA [%-change]
Face split comp.	16.25	+5.8	1.483	+14.60
Interlaced comp.	15.98	+4.1	1.395	+7.8
Face split EXV	12.82	-16.5	0.898	-30.7
Interlaced EXV	15.35	0	1.294	0

Wet conditions

	$\dot{Q}_e$ [kW]	$\dot{Q}_e$ [%-change]	UA [kW/K]	UA [%-change]
Face split comp.	17.66	+2.8	1.816	+9.10
Interlaced comp.	17.39	+1.3	1.707	+2.56
Face split EXV	14.75	-14.1	1.132	-32.0
Interlaced EXV	17.16	0	1.665	0

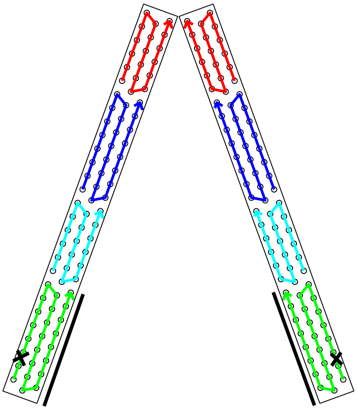
Possible area savings

How can we realize the improvements? (or validate?)

Possible area savings

Face split blockage

- Block of both refrigerant and airflow paths
- While keeping air volume flow constant
- One bottom "face" equals 14% face area
- With compensation in wet conditions



Possible area savings

Face split blockage

	No blockage	14% blockage	28% blockage
Cooling capacity (air-side)	15.56 kW	15.33 kW	14.91 kW
Cooling capacity (ref-side)	16.01 kW	15.61 kW	15.34 kW
Mass flow rate	0.0955 kg/s	0.0930 kg/s	0.0907 kg/s
Evaporator outlet pressure	10.73 bar	10.47 bar	10.28 bar
Sensible heat	12.27 kW	12.16 kW	11.93 kW
Latent heat	3.24 kW	3.12 kW	2.93 kW

Note that the simulations showed only 7-9% UA improvements

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Conclusion

- Hypothesis confirmed by simulations in both dry and wet conditions
  - Face split performs better at compensation in general
- Capacity and UA-value gain is slightly lower in wet conditions
- Benefits in using compensation were validated by possible area savings on the face split evaporator
- It is not always optimal to have equal channel superheats depending on circuitry

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3.6 Engineering the Heat Curve

Ulrik Larsen *(ular@mek.dtu.dk)*
DTU Mechanical Engineering

Timetable ▲
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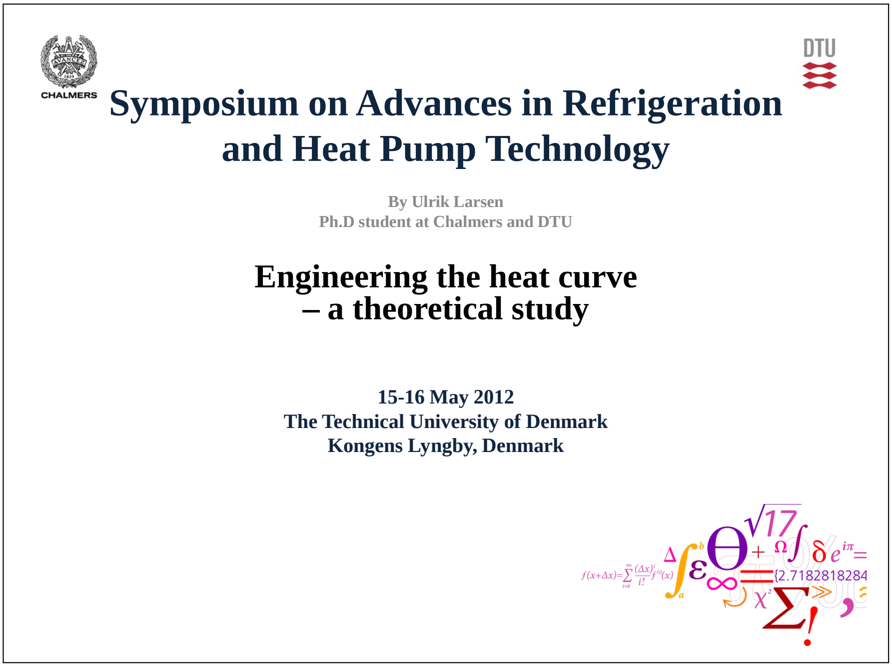


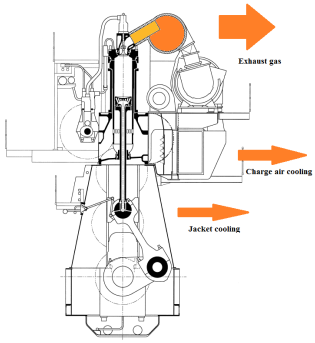
Overview

- Motivation and relevance
- Context of the study
- The Kalina cycle
- Engineering the heat curve
- Model description
- Results

Motivation and relevance

- The motivation for increasing energy efficiencies is widespread
- Many industrial processes take use of evaporators and condensers
- Gliding evaporation temperatures are attracting increasing interest

Context - Propulsion of large ships



Container feeder class

2500 TEU

One large low speed two-stroke main engine
45 kg/s exhaust gas at 240°C at 75% load

Context - Propulsion of large ships

- Looking at two alternatives to the ordinary Rankine cycle for waste heat recovery:
  - Organic Rankine cycle (ORC)
    - Organic working media
    - Relatively simple cycle
  - The Kalina cycle
    - Ammonia-water mixture working media
    - Relatively complicated cycle

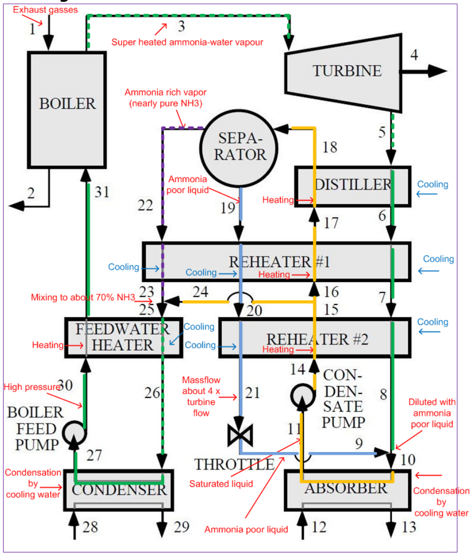
CHALMERS

The Kalina cycle - Intro

DTU

- Named after inventor Dr. Kalina
- New power cycle first published 1983
- Several layouts was presented each for a specific purpose (temperature level)
- Mainly heat from gas turbines, combustion engines, industrial waste heat, geothermal heat and solar power

Alexander Kalina



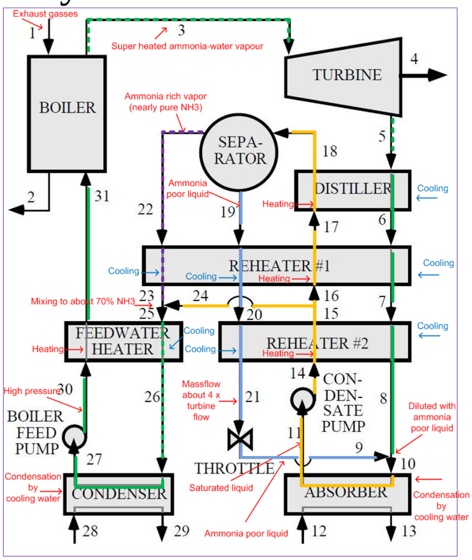
CHALMERS

The Kalina cycle - Intro

DTU

- Two component working media - ammonia and water
- Three pressure levels
- Four concentration levels
- The ‘heart’ of the process is the separator

Alexander Kalina



The Kalina cycle - Binary mixtures

- Composition of working fluid can be varied (using a separator)
- Evaporation and condensation happens at varying temperature

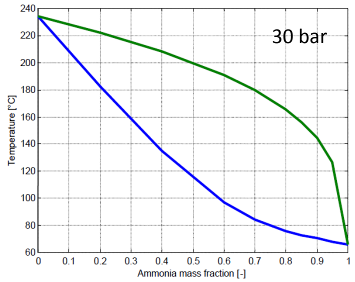


Figure by Páll Valdimarsson

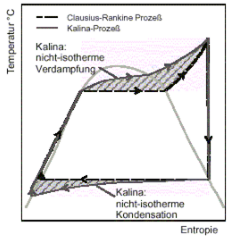


Figure source unknown

The Kalina cycle - Binary mixtures

- The heat curve is similar to one of a super critical pure fluid
- Heat curve for Husavik geothermal heat and power station – Kalina compared to ORC

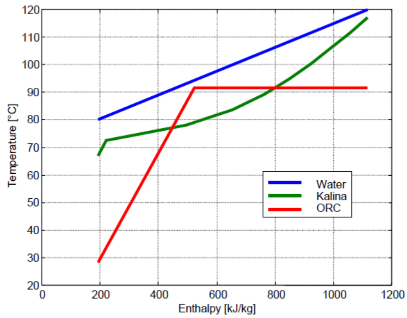
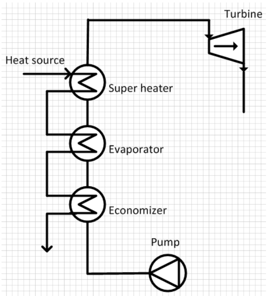
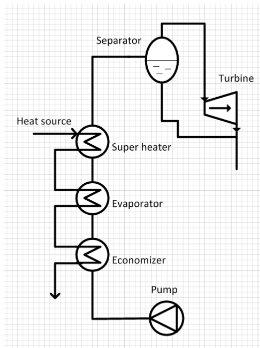


Figure by Páll Valdimarsson

Engineering the heat curve

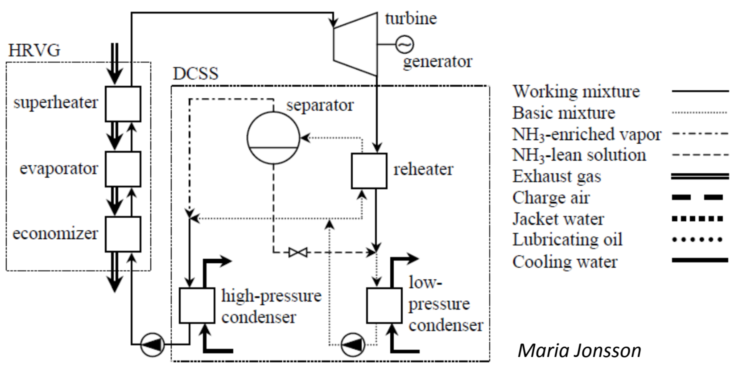
- Increasing the degree of control requires increased process complexity





Engineering the heat curve

- Increasing the degree of control requires increased process complexity

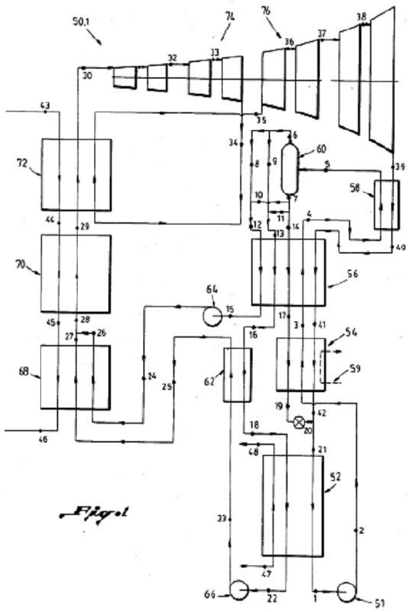


Maria Jonsson

CHALMERS

Engineering the heat curve

- Increasing the degree of control requires increased process complexity – the split cycle Kalina



Alexander Kalina

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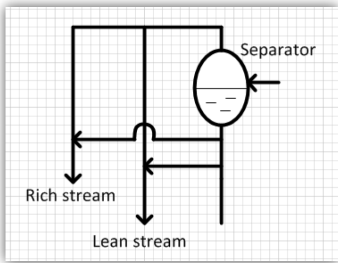
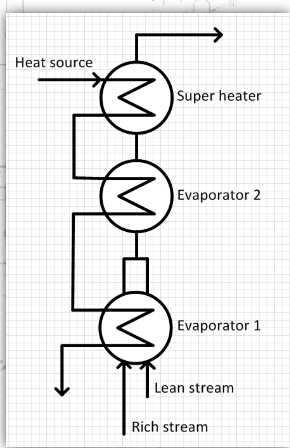
Engineering the heat curve with split cycle

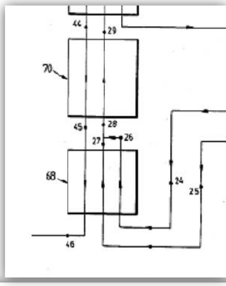
- The two features that makes the split cycle layout possible

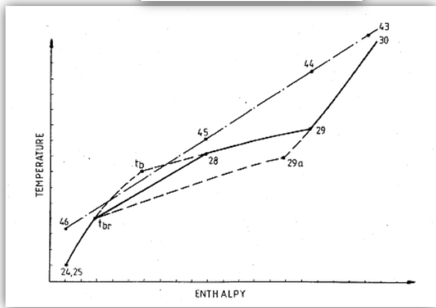




Theoretical heat curve for the split cycle

- The line 24, 25 → t_{br}
  - Both streams are pre-heated
  - At t_{br} the rich stream starts to boil (at a low temperature)
- The line t_{br} → 29a → 30
  - represents the heat curve if only the rich stream was used

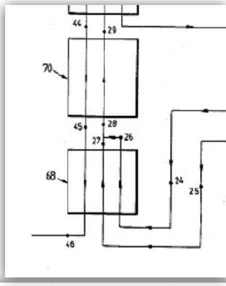


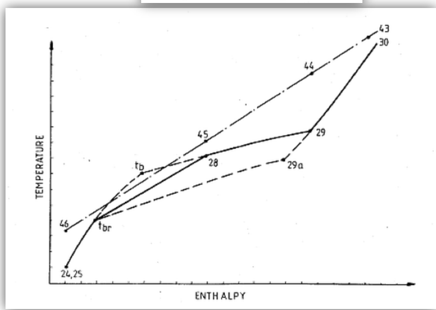


Alexander Kalina

Theoretical heat curve for the split cycle

- The line t_{br} → t_b → 29 → 30
  - represents (while impossible) the heat curve if the two streams where mixed into one stream at point 24, 25
- The curve t_{br} → 28 → 29 → 30
  - Is the heat curve as intended in the split cycle



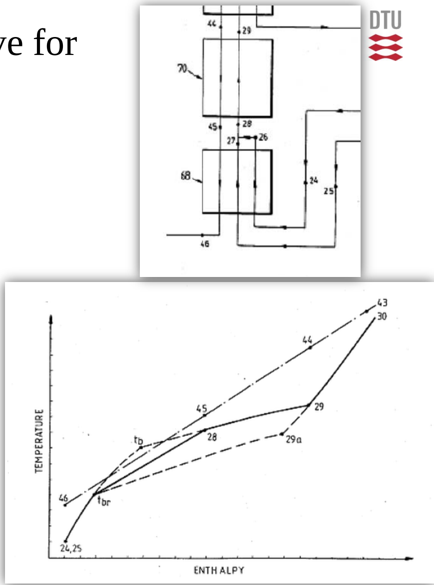


Alexander Kalina

Theoretical heat curve for the split cycle

- Result: The average temperature at which the same amount of heat is transferred to the working fluid is higher

- …so efficiencies are theoretically higher

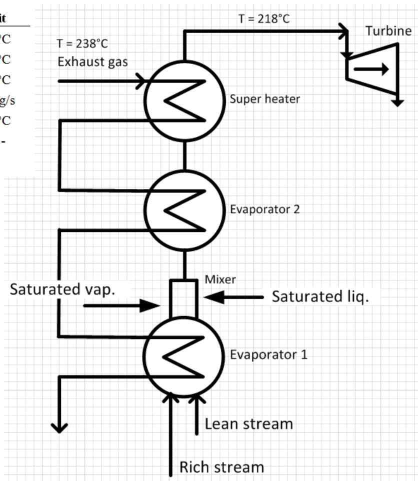


Alexander Kalina

Model description

Property	Value	Unit
Temperature of exhaust gas in	238	°C
Temperature of exhaust gas out	160	°C
Temperature of working media before turbine	218	°C
Mass flow of exhaust gas	47.3	kg/s
Minimum allowable temperature approach in heat exchangers	20	°C
Minimum allowable vapour quality out of turbine	0.90	-
Isentropic/ mechanical efficiency of turbine	0.87/0.98	

- It is assumed that the separator can produce the needed flow of the rich and lean streams



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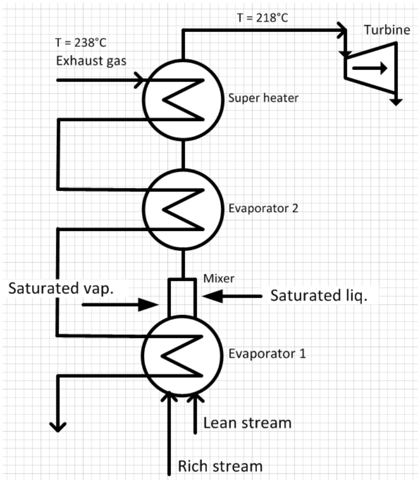
DTU

Model description

- According to Kalina the rich and lean streams should have identical temperatures at the inlet of evaporator 1 and also identical at the outlet
- Same pressure
- This requires Eva 1 outlet conditions (T, xA) where:

  $T_{dew,\ rich} = T_{bubble,\ lean}$



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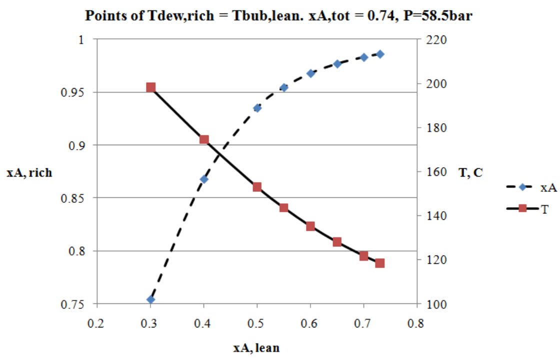
DTU

Model description

- Outlet of EVA 1:

  $T_{dew,\ rich} = T_{bubble,\ lean}$

- The lean and rich concentrations are thus given as a function of T, P and x_A out of EVA
- (or vice versa)



Model description

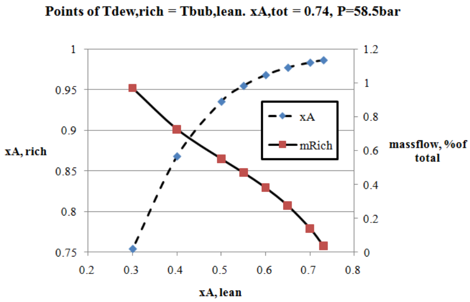
- Having the composite stream ammonia concentration fixed, causes the massflows of lean and rich streams to be given by the ammonia mass balance:

$$x_{A,composite} * \dot{m}_{total} = x_{A,rich} * \dot{m}_{rich} + x_{A,lean} * \dot{m}_{lean}$$

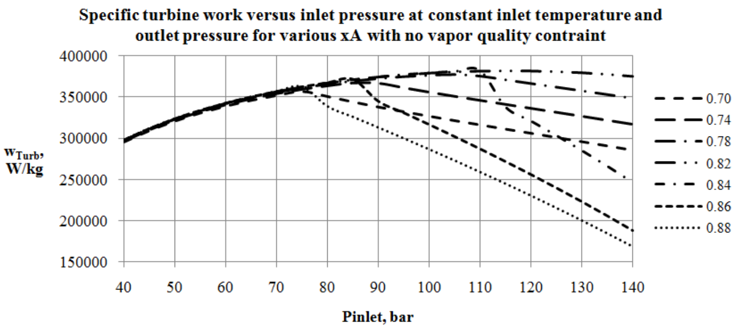
- Thus the massflows are also partially constrained by the T, P and x_A out of EVA 1

- For example when $x_{A,rich}$ is high the massflow_{rich} is low

- These mechanisms constitute the boundaries for engineering the heat curve



Baseline – not using split cycle



- Constant inlet temperature
- The turbine outlet pressure is held constant
  - (normally being a function of the condensing temperature but things are more complicated in the Kalina cycle since the saturation pressure depends on composition)

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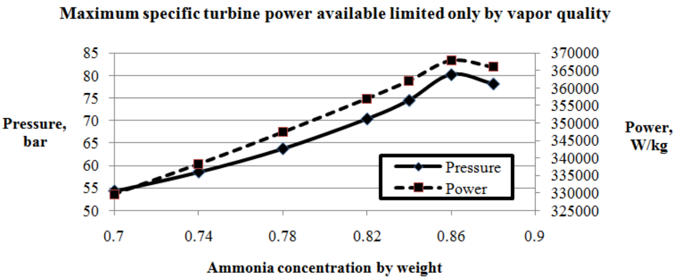
Baseline – not using split cycle

DTU



- Power output per kg working fluid increases with ammonia concentration peaking around 86% by weight

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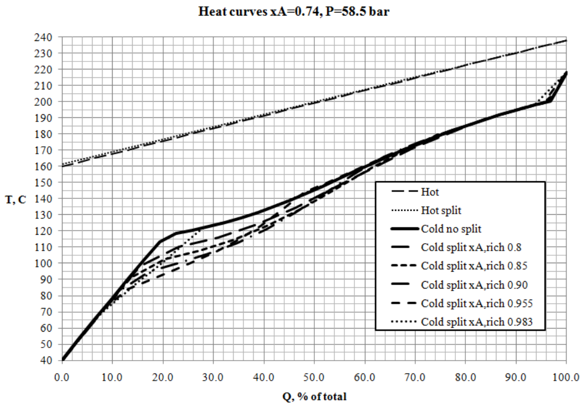
Results

DTU



- First case: Poor recuperation before evaporator; T_{in} = 40°C. No gains using split cycle

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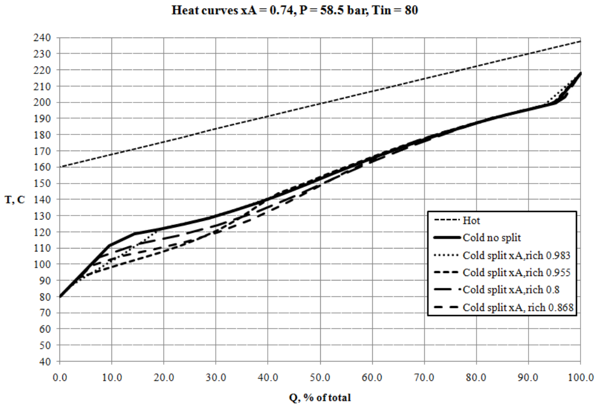
Results



- Second case: Good recuperation before evaporator; T_{in} = 80°C. No significant gains using split cycle

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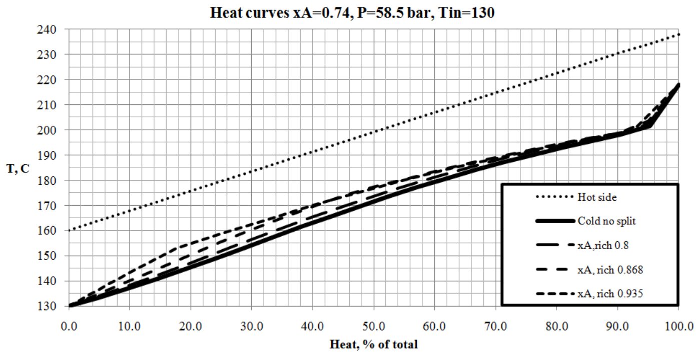
Results



- Third case: Good recuperation and external low temperature heat added before evaporator; T_{in} = 130°C. Clearly the split cycle improves the heat curve.

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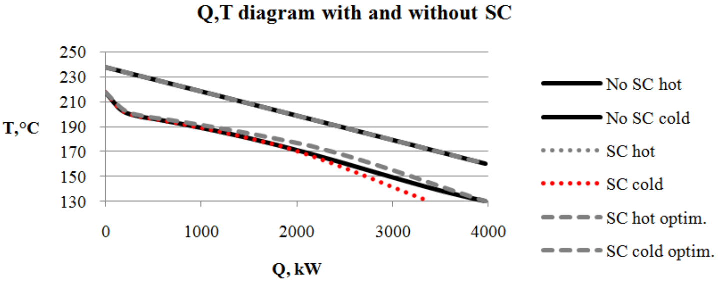
Results

- The improved heat curve enables a higher massflow of working media and thus higher turbine effect (using the same heat source)



CHALMERS

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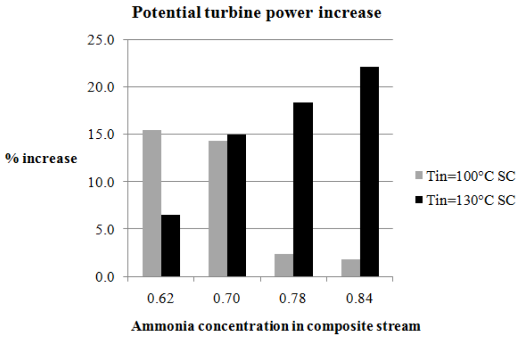
Results



- Various concentrations were investigated – all showing significant potential for increases in energy efficiency using the split cycle

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DTU

Conclusions

- The Kalina split cycle offers the possibility to engineer the heat curve to reduce thermodynamic losses (exergy)
- Under the right conditions the split cycle can provide significantly increased cycle output
- The technique could likely be used in many other applications which has heat intake and rejection
- The technique is likely to offer similar advantages using other suitable working fluid mixtures

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DTU

Thanks – questions?

Go back to the table of contents ⯒ or to the timetable ▲

3.7 Cost and Energy Efficiency of Air-Water Heat Pumps

Gunda Mader *(gunda.mader@danfoss.com)*
Danfoss A/S

Timetable ▲
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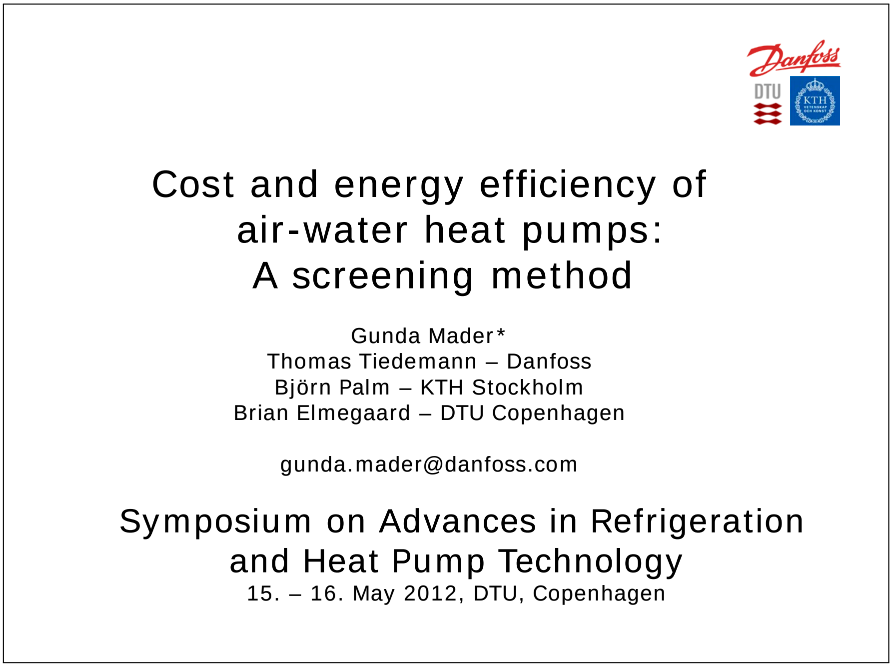

Content

- » Improving a heat pump
- » Problem setup
- » Challenges
- » Screening method
  - » Technology selection & ideal control
  - » Design of experiment
  - » Optimization
- » Exemplary results
- » Conclusions

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Improving a heat pump

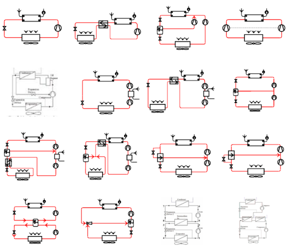
» Refrigerants

R717
E170
R152a
R32
R134a
R1270
R290
R407C
R410A
R423A
R143a
R404A
R507A
R125
R218
R1234yz

» Components

- » Heat exchangers:
  - » geometry, size
- » Expansion devices
- » Compressors:
  - » type, size, efficiency curves
- » Fans, valves, receivers …

» Cycles



» Control

- » Capacity control
- » Superheat control
- » Fan control
- » Cycle control on advanced cycles
- » Defrost control

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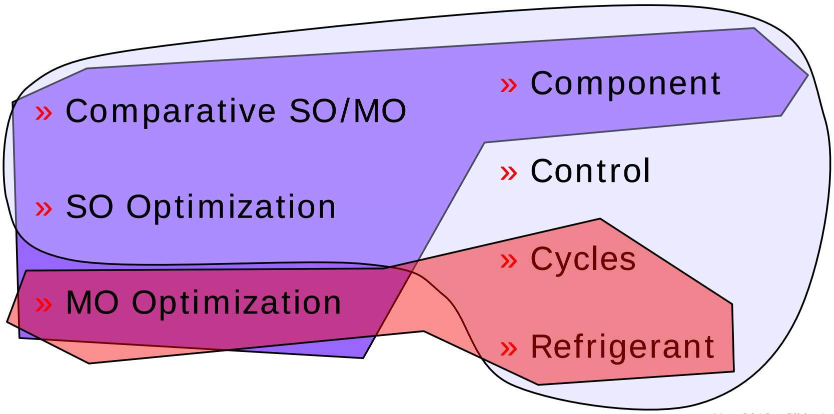
Improving a heat pump

Objectives:

Energy efficiency, Costs, Safety, Reliability, Noise ...



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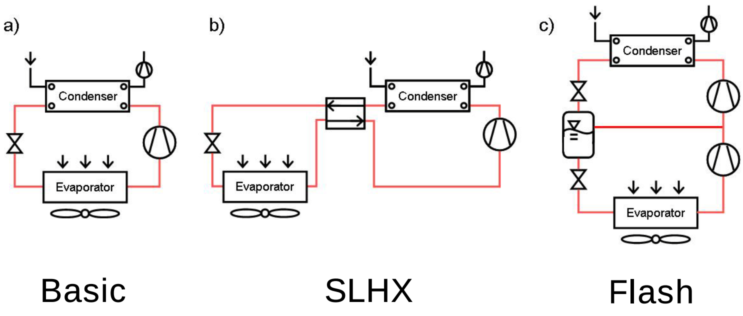
Problem setup

Exemplary selection



Refrigerants: R410A, Propane

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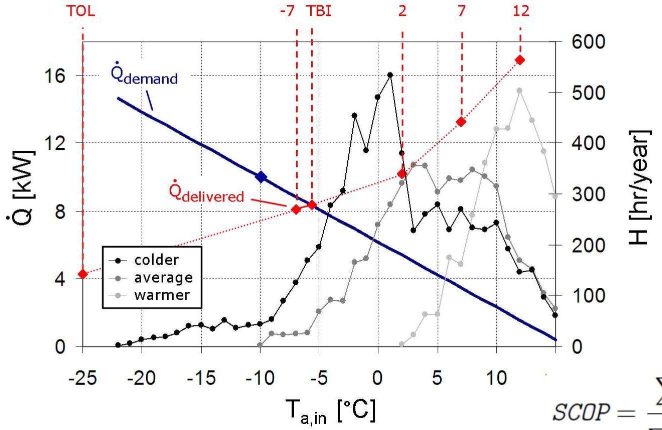
Problem setup

1. Objective function: SCOP



EN 14825

$$SCOP = \frac{\sum_j H_j \dot{Q}_j}{\sum_j H_j \frac{\dot{Q}_j}{COP_j}}$$

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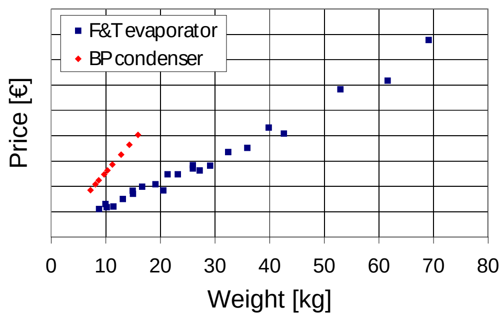
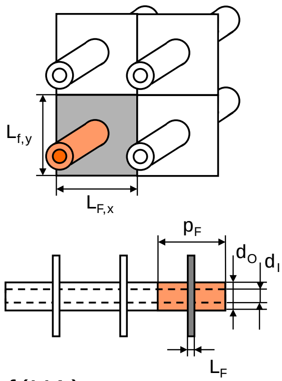
Problem setup

2. Objective function: Costs





$Cost_{HX}$ = f(heat exchange area)$_{HX}$ = f(UA)$_{HX}$
$Cost_{other}$ = f(technology)

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Challenge: Modeling

How to model components & control when comparing cycles?

- Increasing level of detail
  - increases number of variables
  - increases number of assumptions
  - complicates evaluation
- Need to reflect dominating physical behavior while keeping models simple

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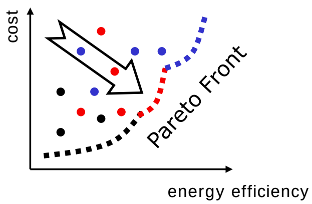
Challenge: MO Optimization

- Non-linear mixed integer problem
- Evolutionary algorithms



  - numerical problems
  - (extremely) high computation time
  - difficult result evaluation
  - coupling of objectives

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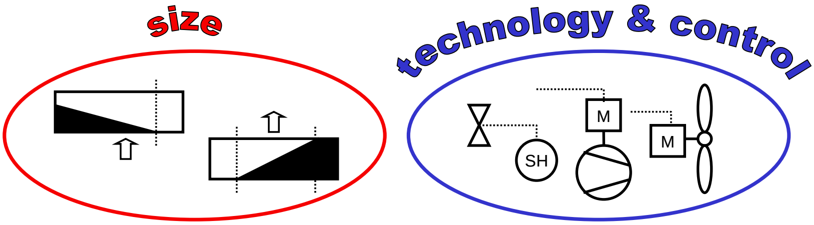
Screening method



Continuous variables
Design of Experiment
& Quadratic model

Integer variables
Technology selection
& Ideal control

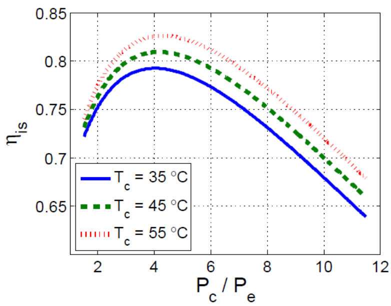
Optimization

Sensitivity Analysis

May 2012 - Slide 10

Ideal control

» expansion device
  » $h_{in} = h_{out}$
  » superheat: 2 K
» compressor
  » reciprocating



  » capacity control: optimal demand match (idealized variable speed)

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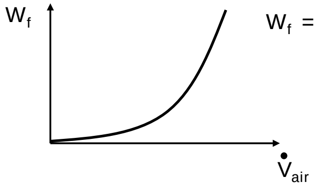
Ideal control

» air flow rate: optimized for operating condition (max. COP)

» assumption of pressure curve required



$W_f = f_f \dot{V}_{air}$

$\dot{V}_{air} \uparrow \implies W_f \uparrow, P_e \uparrow$

» subcooling: optimized for operating condition (max. COP)

$SC \uparrow \implies P_c \uparrow, m_{ref} \downarrow$

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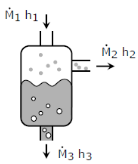
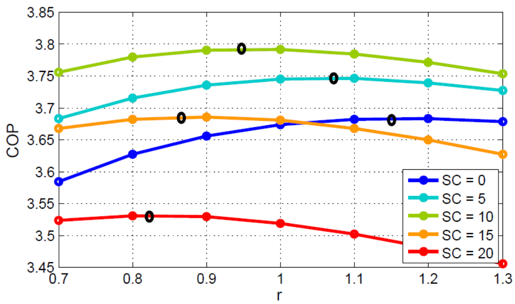
Ideal control (Flash cycle)



» ideal phase separation

» intermediate pressure level: $r = \dfrac{P_m}{\sqrt{P_{suc}P_{dis}}}$



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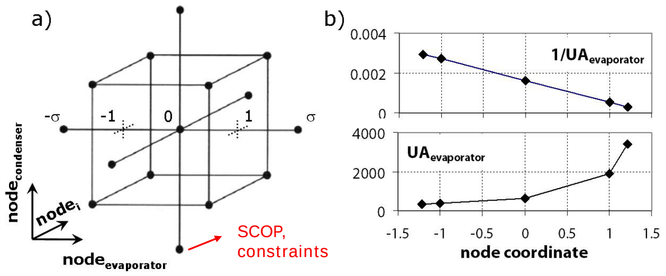
Design of Experiments

Decision variable: heat exchange area / UA value



Quadratic model: SCOP = f(UAe, UAc, ...)
+ constraints = f(UAe, UAc, ...)

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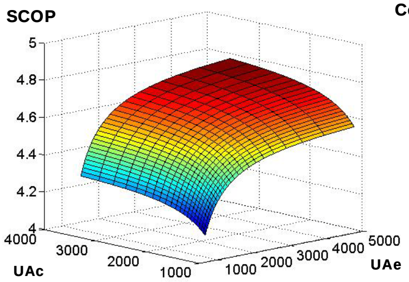
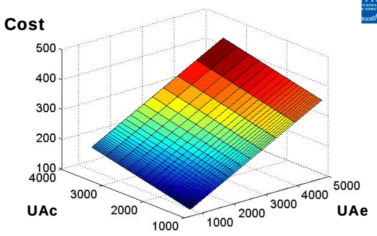
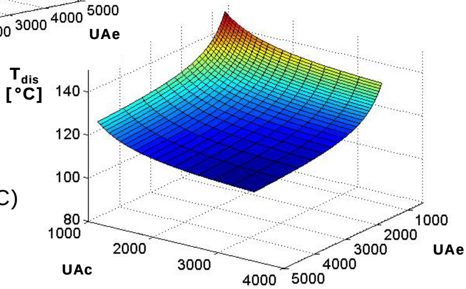
Quadratic models & cost function

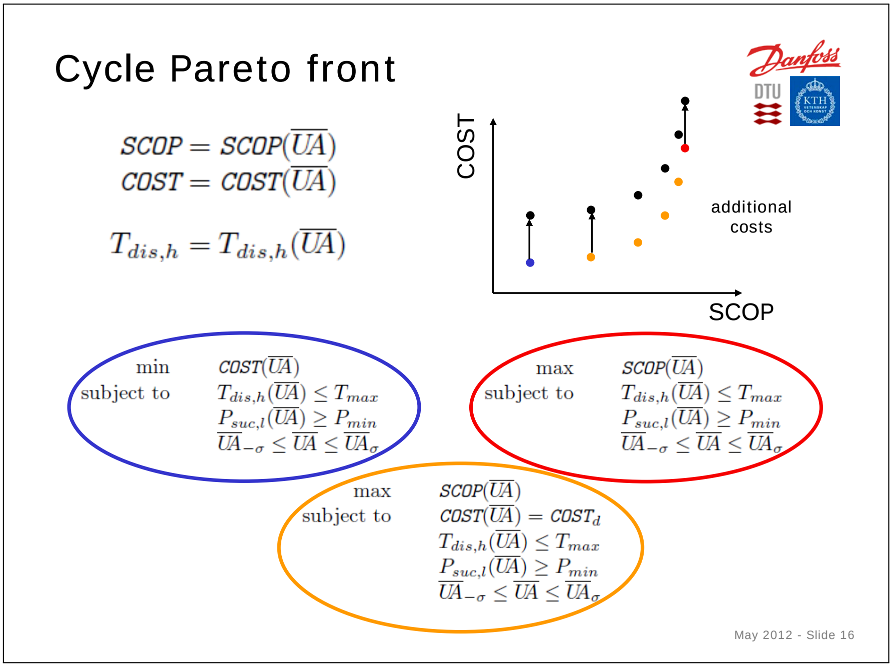






»Basic cycle
»R410A
»Medium T_{wout} profile(45° C)
»Cold climate

Slide 15


Cycle Pareto front
$SCOP = SCOP(\overline{UA})$
$COST = COST(\overline{UA})$
$T_{dis,h} = T_{dis,h}(\overline{UA})$
COST
SCOP
additional costs
min $COST(\overline{UA})$
subject to $T_{dis,h}(\overline{UA}) \leq T_{max}$
$P_{suc,l}(\overline{UA}) \geq P_{min}$
$\overline{UA}_{-\sigma} \leq \overline{UA} \leq \overline{UA}_{\sigma}$
max $SCOP(\overline{UA})$
subject to $T_{dis,h}(\overline{UA}) \leq T_{max}$
$P_{suc,l}(\overline{UA}) \geq P_{min}$
$\overline{UA}_{-\sigma} \leq \overline{UA} \leq \overline{UA}_{\sigma}$
max $SCOP(\overline{UA})$
subject to $COST(\overline{UA}) = COST_d$
$T_{dis,h}(\overline{UA}) \leq T_{max}$
$P_{suc,l}(\overline{UA}) \geq P_{min}$
$\overline{UA}_{-\sigma} \leq \overline{UA} \leq \overline{UA}_{\sigma}$
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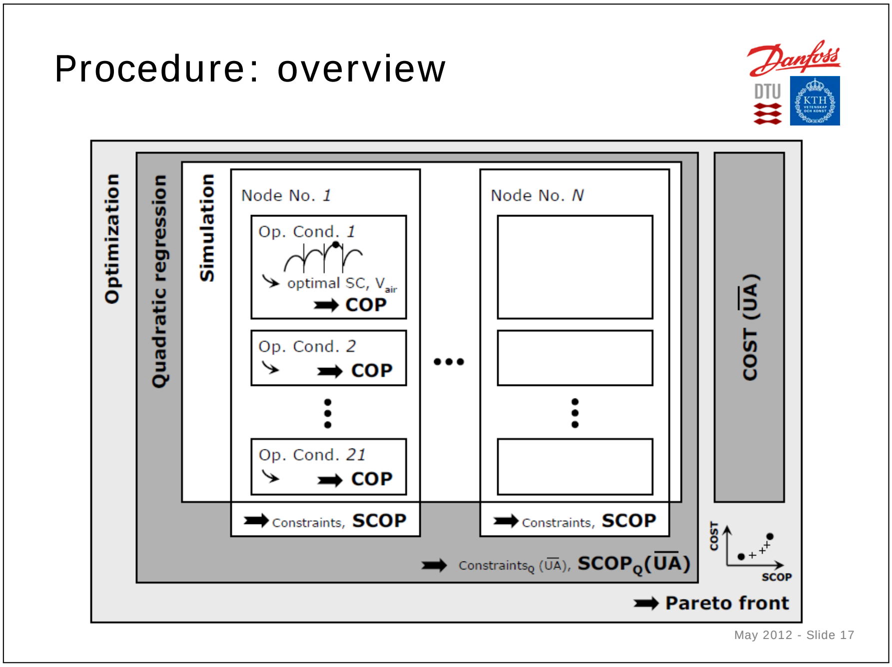
Procedure: overview
Optimization
Quadratic regression
Simulation
Node No. 1
Op. Cond. 1
optimal SC, V_{air}
COP
Op. Cond. 2
COP
Op. Cond. 21
COP
Constraints, SCOP
Node No. N
Constraints, SCOP
Constraints$_Q$ ($\overline{UA}$), SCOP$_Q$($\overline{UA}$)
COST ($\overline{UA}$)
COST
SCOP
Pareto front
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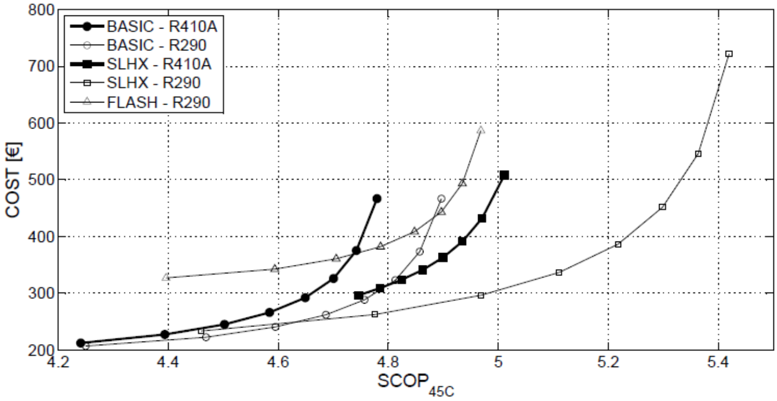
Exemplary result



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Conclusions

Screening method to compare different technologies, cycle layouts and refrigerants in multiple objectives

+ small number of simulations per Pareto front
+ numerical problems have only "local" effects
+ easy integration of constraints
+ decoupled objective functions
+ allows detailed sensitivity analysis

- numerous curves for high number of different technologies
- small error induced by quadratic model (< 1% 2D, < 2% 3D)
- strong component effects might result in broad bands

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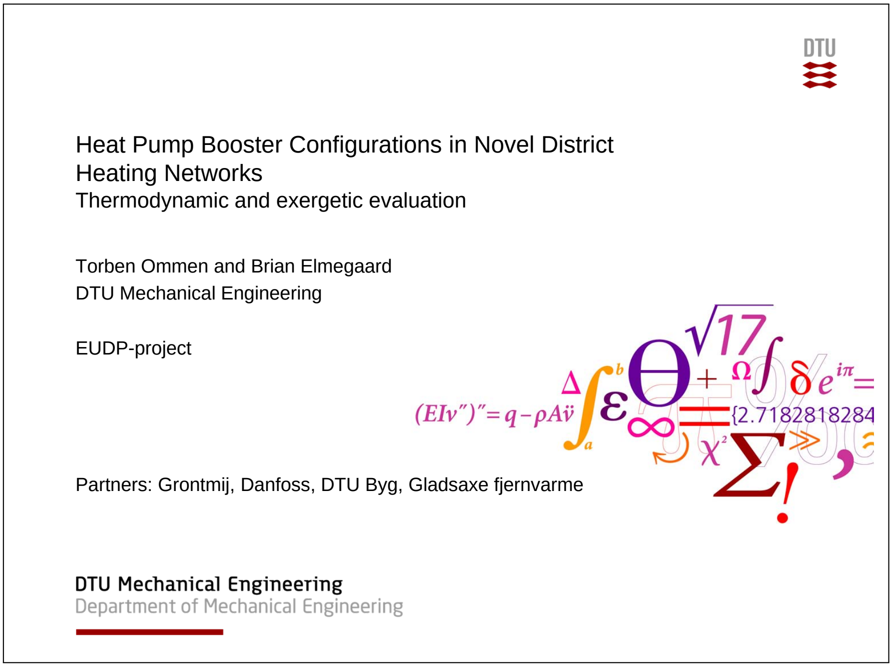
3.8 Heat Pump Booster Configurations in Novel District Heating Networks

Torben Ommen *(tsom@mek.dtu.dk)*
DTU Mechanical Engineering

Timetable ▲
Table of contents ⏶



Project arguments

- Space heating is very fossil fuel intensive, and a large share of Danish consumption.
  - Power and heat production must be covered by renewables in 2035
- Novel floor heating systems allows for lower temperatures in district heating network, as 35-45 °C is sufficiently high temperature for space heating
- Low supply temperature prompts new possibilities in heat production:
  - Waste heat from various sources (e.g. industry)
  - Sustainable heat sources: (Geothermal heat production & Solar collectors etc.)
  - Increased capacity of traditional DH-networks by lowering return temperature
  - Increased power production in
    a) Greenfield CHP projects
    b) DH networks where temperature levels can be changed
  - Lower losses in DH-networks
- Problematic area when using low temperature district heating: Domestic **hot** water

2 DTU Mechanical Engineering, Technical University of Denmark 22 May 2012

Heat Pump Booster Configurations in Novel District Heating Networks

If supply temperature between 35-45 °C:
Challenge to satisfy the domestic hot water demand.

- DS 439 (Code of Practice) assuming ground water temperature 10 °C:
  - High heating demands can occur (32,3 kW)
  - 45 °C tap water (cleaning)
  - Required operating temperature in water heaters: 55 °C
- Solution:
  - Temperature boost required (possible with heat pumps)
  - Stratified storage tank to average the heating demand

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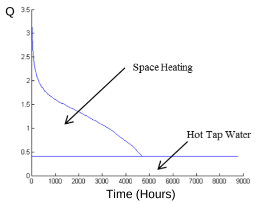
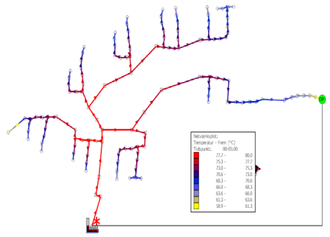
Heat Pump Booster Configurations in Novel District Heating Networks

Case: 116 family houses under construction
- Length of network (3.6 km), ground temperature 10 ºC
- Annual consumption of heat in 159 m2 BR 10 class 2015 house
  - 4040 kWh for space heating
  - 3200 kWh for hot tap water





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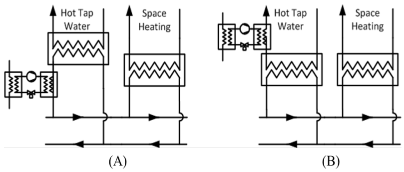
Basic Concepts of Hot Water Supply in LTDH

Basic Concepts:
- Conventional system 80/40 ºC
- Low temperature system 45/25 ºC
  - with electric heating
  - with heat pump and secondary side tank
  - with heat pump and secondary side tank and preheating
  - with heat pump and primary side tank

- Heat demand average
- Space heating 30/22 ºC
- Hot tap water 50/10 ºC

Two different implementation schemes:



(A) Heat pump on primary side of the tap water heat exchanger.
(B) Heat pump on secondary side of the tap water heat exchanger.

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Basic Concepts of Hot Water Supply in LTDH

Assumptions:

Estimated heat loss coefficient per unit pipe length: 65 W/km ºC .

Variable	Assumption
Minimum temperature difference in heat exchanger in network	5 [K]
HEX pinch temperature difference in both Condenser and Evaporator	2.5 [K]
Refrigerant	R134a
Isentropic efficiency of compressor	0.5 [/]
Hot tap water	50 [ºC]
Tap water in	10 [ºC]
Minimum temperature if water stored on secondary side	60 [ºC]
Pressure loss in system	0 [kPa]

Heat loss is only considered in the distribution network

Results

All results are presented in terms of Energy and Exergy

- Changes in kinetic and potential energy are neglected and chemical reactions do not occur, only the physical exergy of the flows are calculated.
- Exergy:

$$\dot{E}_i^{PH} = \dot{m}_i((h_i - h_0) - T_0(s_i - s_0))$$

- Dead state: P_0 = 101 [kPa], T_0=10 [K]

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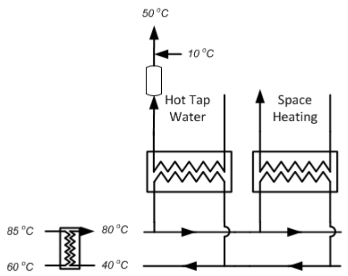
Basic Concepts of Hot Water Supply in LTDH

- Conventional System

		Energy [W]	Exergy [W]
Hot Water	Hot water supply	370	23
	Heat loss	80	13
	District heat consumption	450	81
	Electricity consumption	0	0
	Efficiency [%]	**82**	**29**
Space heating	Heat supply	460	25
	Heat loss	110	17
	District heat consumption	570	102
	Efficiency [%]	**81**	**24**
Total	Heat supply	830	48
	Heat loss	190	30
	Heat consumption	1020	180
	Electricity consumption	0	0
	Efficiency [%]	**81**	**27**

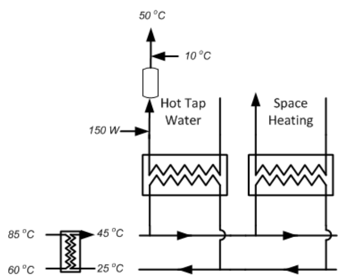


DTU

Basic Concepts of Hot Water Supply in LTDH

- Low temperature system 45/25 °C with electric heating

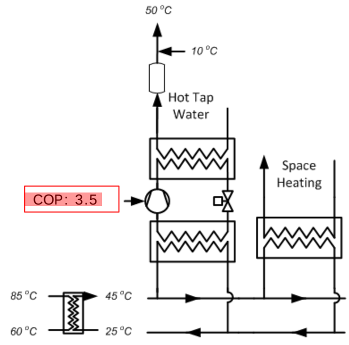
		Energy [W]	Exergy [W]
Hot Water	Hot water supply	370	23
	Heat loss	40	3.7
	District heat consumption	260	47
	Electricity consumption	150	150
	Efficiency [%]	**89**	**12**
Space heating	Heat supply	460	25
	Heat loss	74	9.2
	District heat consumption	530	96
	Efficiency [%]	**89**	**26**
Total	Heat supply	830	48
	Heat loss	180	13
	Heat consumption	790	140
	Electricity consumption	150	150
	Efficiency [%]	**89**	**16**



Basic Concepts of Hot Water Supply in LTDH

- Low temperature system 45/25 ºC with R134a Heat pump

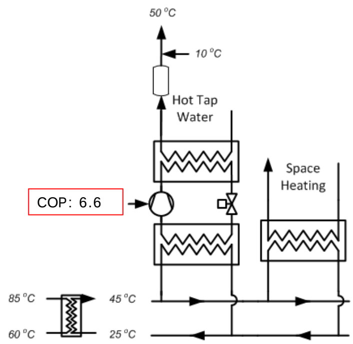
		Energy [W]	Exergy [W]
Hot Water	Hot water supply	370	23
	Heat loss	40	3.7
	District heat consumption	300	55
	Electricity consumption	100	100
	Efficiency [%]	**90**	**15**
Space heating	Heat supply	460	25
	Heat loss	74	9.2
	District heat consumption	530	96
	Efficiency [%]	**89**	**26**
Total	Heat supply	830	48
	Heat loss	110	13
	Heat consumption	840	150
	Electricity consumption	100	100
	Efficiency [%]	**89**	**19**



Basic Concepts of Hot Water Supply in LTDH

- Low temperature system 45/25 ºC with R744 Heat pump

		Energy [W]	Exergy [W]
Hot Water	Hot water supply	370	23
	Heat loss	40	3.7
	District heat consumption	350	63
	Electricity consumption	56	56
	Efficiency [%]	**90**	**20**
Space heating	Heat supply	460	25
	Heat loss	74	9.2
	District heat consumption	530	96
	Efficiency [%]	**89**	**26**
Total	Heat supply	830	48
	Heat loss	110	13
	Heat consumption	880	140
	Electricity consumption	56	56
	Efficiency [%]	**88**	**24**



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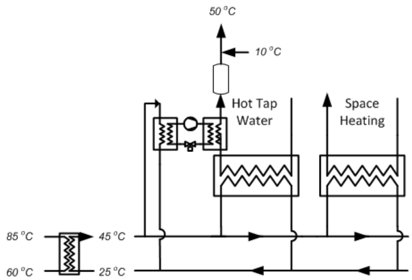
Basic Concepts of Hot Water Supply in LTDH

- Low temperature system 45/25 ºC with heat pump and preheating

		Energy [W]	Exergy [W]
Hot Water	Hot water supply	365	23
	Heat loss	40	3.7
	District heat consumption	360	66
	Electricity consumption	41	41
	Efficiency [%]	**90**	**22**
Space heating	Heat supply	460	25
	Heat loss	74	9.2
	District heat consumption	530	96
	Efficiency [%]	**89**	**26**
Total	Heat supply	830	48
	Heat loss	120	14
	Heat consumption	890	160
	Electricity consumption	41	41
	Efficiency [%]	**89**	**24**



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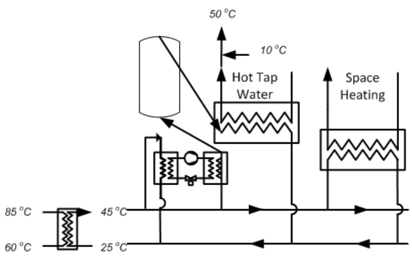
Basic Concepts of Hot Water Supply in LTDH

- Low temperature system 45/25 ºC with heat pump and preheating
  - Tank on primary side

		Energy [W]	Exergy [W]
Hot Water	Hot water supply	370	23
	Heat loss	40	3.7
	District heat consumption	390	68
	Electricity consumption	30	30
	Efficiency [%]	**90**	**22**
Space heating	Heat supply	460	25
	Heat loss	74	9.2
	District heat consumption	530	96
	Efficiency [%]	**89**	**26**
Total	Heat supply	830	48
	Heat loss	120	14
	Heat consumption	900	160
	Electricity consumption	30	30
	Efficiency [%]	**89**	**25**



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Basic Concepts of Hot Water Supply in LTDH

Summery of Basic Concepts:

System	Distribution temperatures [°C]	Refrigerant	Heat pump COP [–]	Tank location	Preheating	Heat consumption [W]	Electricity consumption [W]	Energy Efficiency [%]	Exergetic Efficiency [%]
Conv. 1 (3.1.1)	80/40	–	–	Sec		1000	0	81	27
LT EL (3.2)	45/25	–	1.0	Sec		790	150	89	16
LT HP 1 (3.3.1)	45/25	R134a	3.5	Sec		830	100	88	19
LT HP 2 (3.3.2)	45/25	R744	6.6	Sec		880	56	88	24
LT HP 3 (3.4)	45/25	R134a	3.5	Sec	×	890	41	89	24
LT HP 4 (3.5)	45/25	R134a	4.0	Prim	×	900	30	89	25

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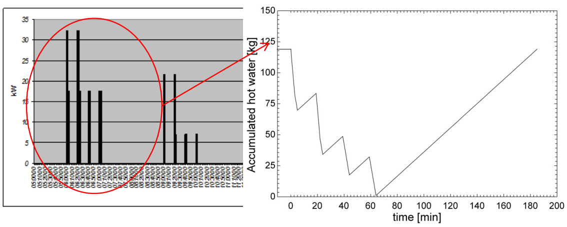
Storage Tank requirement and configurations

- The storrage tank is required as large district heating network has small temperature difference between supply and return line.
- To avoid:
  - High massflows of district heating water
  - Large piping and large heat losses
  - High pressure drops



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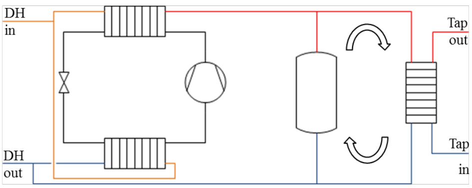
Heat Pump Booster Configurations in Novel District Heating Networks

- Evaluation of configurations with focus on practical considerations.
  - Only the best configurations are taken from the 'basic' evaluation

Variable	Assumption
Pinch temperature in Tap-water HEX (Q_{MAX}=32 kW)	8 [K]
Initially assumed forward temperature of DH network	40 [ºC]
Initially assumed return temperature of DH network	22 [ºC]
Refrigerant	R134a
Isentropic efficiency of compressor	0.5 [/]
HEX pinch temperature difference in both Condenser and Evaporator	2.5 [K]
Hot tap water	45 [ºC]
Tap water in	10 [ºC]
Minimum temperature if water stored on secondary side	58 [ºC]

Configuration A



Variant	$\dot{V}_{DH}$ [m³/h]	Condenser [kW]	P [kW]	Heat pump COP [/]	Water Volume [m³]	Exergetic eff. [/]
A	0.107	0.89	0.157	5.62	0.118	0.44

Volumeflow of 80/40 system during 32 kW peak (without storage): 0,7 [m³/h]

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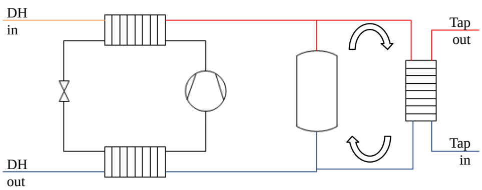
Configuration B



Variant	$\dot{V}_{DH}$ [m³/h]	Condenser [kW]	P [kW]	Heat pump COP [/]	Water Volume [m³]	Exergetic eff. [/]
B	0.059	0.89	0.252	3.52	0.118	0.38

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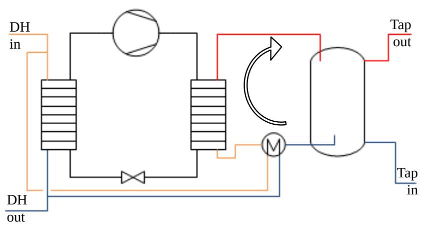
Configuration C



Variant	$\dot{V}_{DH}$ [m³/h]	Condenser [kW]	P [kW]	Heat pump COP [/]	Water Volume [L]	Exergetic eff. [/]
C	0.105	1.02	0.193	5.26	0.086	0.40

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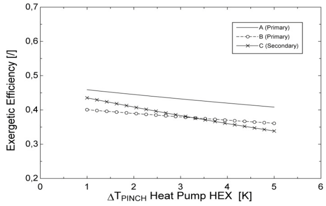
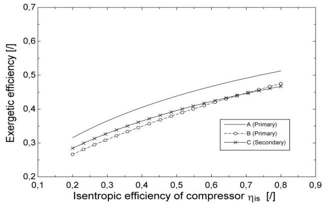
Comparison of investigated configurations

Variant	$\dot{V}_{DH}$ [m³/h]	Condenser [kW]	P [kW]	Heat pump COP [/]	Water Volume [m³]	Exergetic eff. [/]
A	0.107	0.89	0.157	5.62	0.118	0.44
B	0.059	0.89	0.252	3.52	0.118	0.38
C	0.105	1.02	0.193	5.26	0.086	0.40



Impact of pinch temperature difference in evaporator and condenser on the 3 proposed configurations.



Impact of variable isentropic efficiency on the 3 proposed configurations

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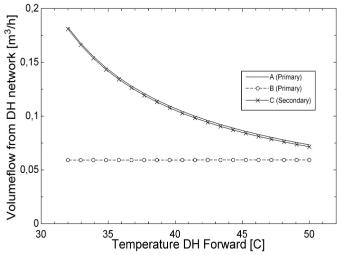
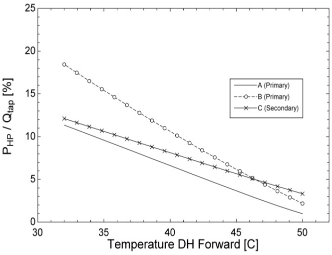
Variable Temperature levels of DH-network - Supply temperature

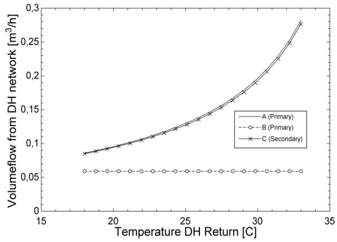


Required volume flow of hot DH stream with variable forward DH temperature.

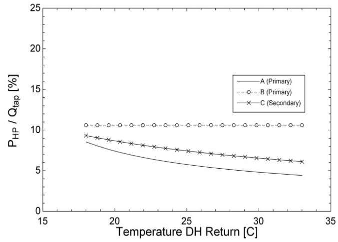


Relation between electricity consumption and product with variable forward DH temperature

Variable Temperature levels of DH-network - Return temperature

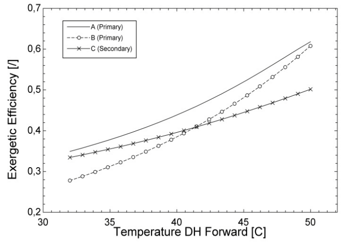


Required volume flow of hot DH stream with variable forward DH temperature.

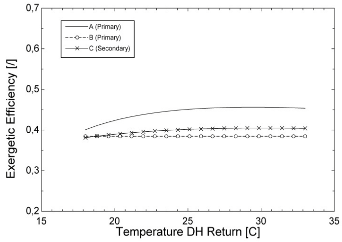


Relation between electricity consumption and product with variable return DH temperature

Variable Temperature levels of DH-network – Exergetic evaluation

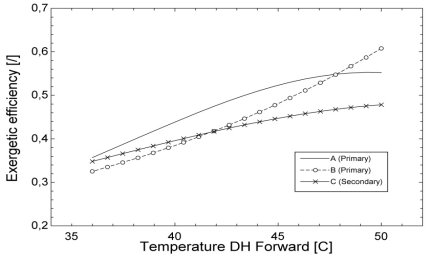


Required volume flow of hot DH stream with variable forward DH temperature.



Relation between electricity consumption and product with variable forward DH temperature

Variable Temperature levels of DH-network – Constant temperature difference between supply and return



Exergetic efficiency of the individual configurations with constant temperature difference between forward and return line of the district heating network.

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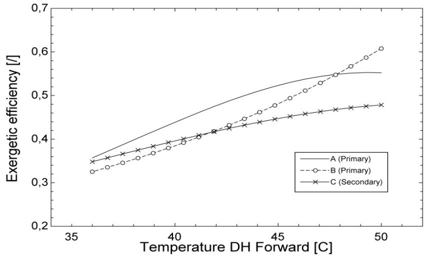
Variable Temperature levels of DH-network – Constant temperature difference between supply and return



Exergetic efficiency of the individual configurations with constant temperature difference between forward and return line of the district heating network.

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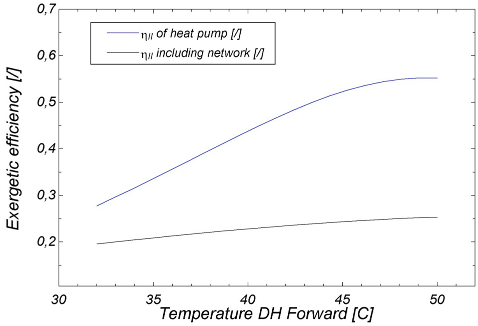
Variable Temperature levels of DH-network – Constant temperature difference between supply and return with heat losses in DH-network



Exergetic efficiency of the individual configurations with constant temperature difference between forward and return line of the district heating network – with heat losses in the network

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Practical barriers in development & of the heat pump

- Requirements for the heat pump unit
  - Easy integration in the district heating system
  - Robust design
  - Increased energy efficiency (not further specified)
  - Better economy in the combined DH-network

  - Natural refrigerant
  - High COP
  - Small size

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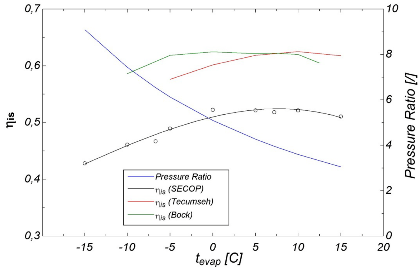
Five heat pump units as demonstration

Assuming isentropic efficiency 0.5:

Variant	$\dot{V}_{DH}$ [m³/h]	Condenser [kW]	P [kW]	Heat pump COP [/]	Water Volume [m³]	Exergetic eff. [/]
A	0.107	0.89	0.157	5.62	0.118	0.44

- In reality (R600a):



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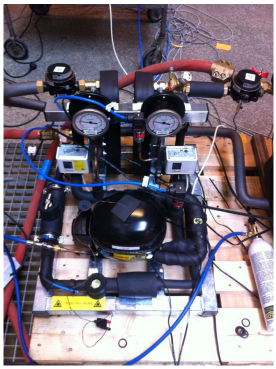
Experimental investigations



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Conclusion

- Investigation of heat pumps implementations in Low Temperature District Heating were supply is between 32 [ºC] and 52 [ºC]

- 5 different configurations have been investigated in the initial analysis of the system
  - Second-law efficiency between 20-30 %
  - Traditional solution more efficient, but only from DH-station and forward
  - HP on primary side most efficient, secondary side OK with preheating or R744

- 3 configurations were evaluated in further analysis
  - Second law efficiency between 30-60 % in the in-house installations
  - Primary side solution most efficient
  - Volume flow rates acceptable for all cases using stratified tank

- Experimental setup being tested in Aarhus, Refrigerant: R600a

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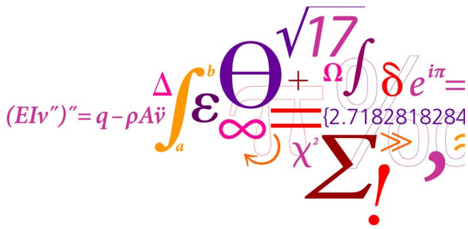
Heat Pump Booster Configurations in Novel District Heating Networks
Thermodynamic and exergetic evaluation

Torben Ommen and Brian Elmegaard
DTU Mechanical Engineering

EUDP-project



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Department of Mechanical Engineering

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3.9 Application of Industrial Heat Pumps

Svend V. Pedersen *(svp@teknologisk.dk)*
DTI

Timetable ▲
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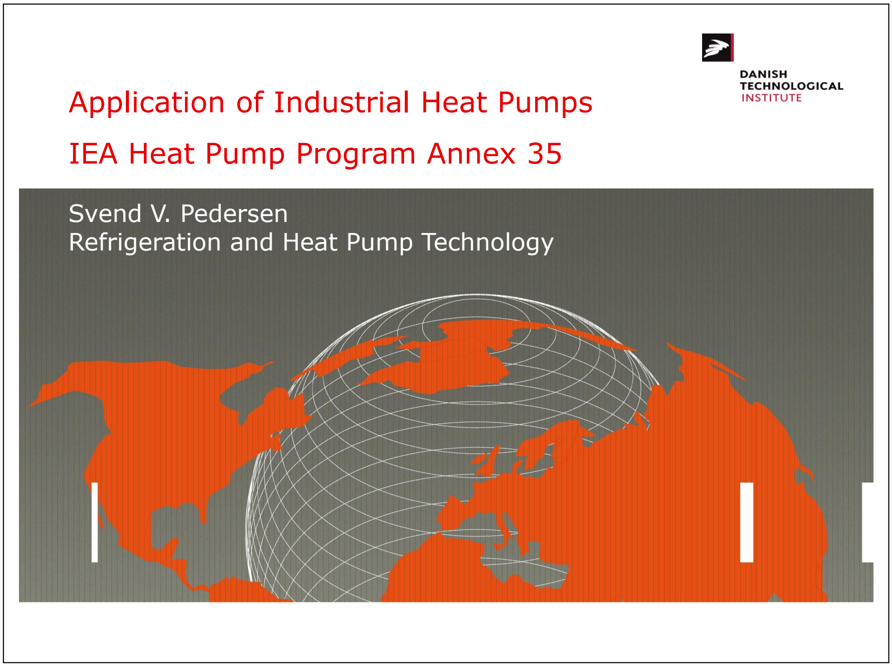

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Contents

- **Description of the project; Objectives/Participants**
- **Contents of the project:**
  Task 1: Market overview, barriers for application
  Task 2: Modeling calculation and economic models
  Task 3: Technology high-temperature heat pumps, process technological integration, refrigerants
  Task 4: Application and monitoring, easy to install standard solutions, operating experience energy effects
  Task 5: Communication
  awareness of potential (policy paper)
  internet, database , training.

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Objectives of the Annex

Based on task-sharing activities:
- Gathering information from experience in running projects.
- Setting up and monitoring demonstration projects or field experiments.
- Publishing and evaluating the results as an information source.
- Giving guidelines to new developments and starting of new (collective) projects.
- Holding regular workshops.

Duration:
- Starts 2010 and ends in 2013.

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Participants

- **Austria** TU Graz, AIT Austrian Institute of Technology
- **Canada** Canmet Energy, Hydro-Quebec Research Institute Laboratoire des technologies de l'énergie (LTE)
- **Germany** Information Centre on Heat Pumps and Refrigeration IZW e.V. Institut für Energiewirtschaft & Rationelle Energieverwendung IER, Universität Stuttgart, Thermea.energiesysteme GmbH, Freital, Vilter Emmerson
- **Netherlands** ECN Efficiency & Infrastructure
- **France** Program Research and Innovation of ECLEER-Industry EDF-R&D-ECLEER
- **Japan** Heat Pump & Thermal Storage Technology Center of Japan, Technical Research Department
- **South Korea** Solar Thermal and Geothermal Research Center New and Renewable Energy Research Department Korea, Institute of Energy Research
- **Sweden** SP Technical Research Institute of Sweden, Energy Technology, KTH Royal Institute of Technology, Chalmers University
- **Switzerland** KWT Kälte Wärmetechnik AG, Viessmann Group, Ecole Polytechnique Fédérale de Lausanne (EPFL), Laboratoire d'Energétique Industrielle (LENI),
- **Denmark** Danish Technological Institute, Advansor, Grontmij Carl Bro, Cool Partners

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Annex – Definition "Industrial Heat Pumps"

- **Heat pumps in the medium and high power ranges that can be used not only for heat recovery in industrial processes but also for heating, cooling and air-conditioning in commercial, and industrial buildings as well as in district heating.**

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Task 1: Market overview

- **Country reports on:**

- Overview of the energy situation in participating countries
- Overview of energy use in segments of industries in participating countries
- Overview of the energy use for heating and cooling in industrial, commercial and large residential buildings.

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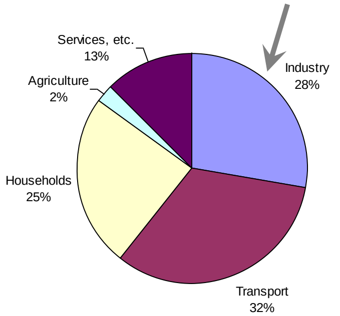
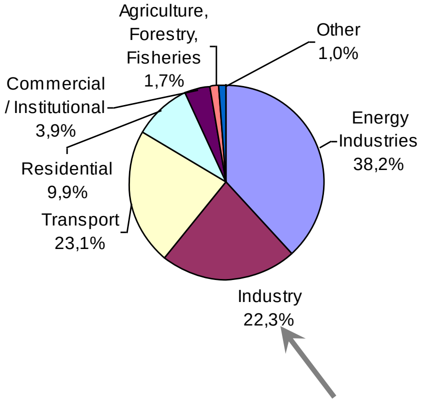
Task 1: Market overview, Potentials

Final Energy Consumption -EU 27- by Sector (2007)



EU27 final energy consumption by sectors 2007 (Mtoe)

EU27 CO_2 Emissions by sector 2007 (Mt)



Energy use in the Danish manufacturing industry based on processes

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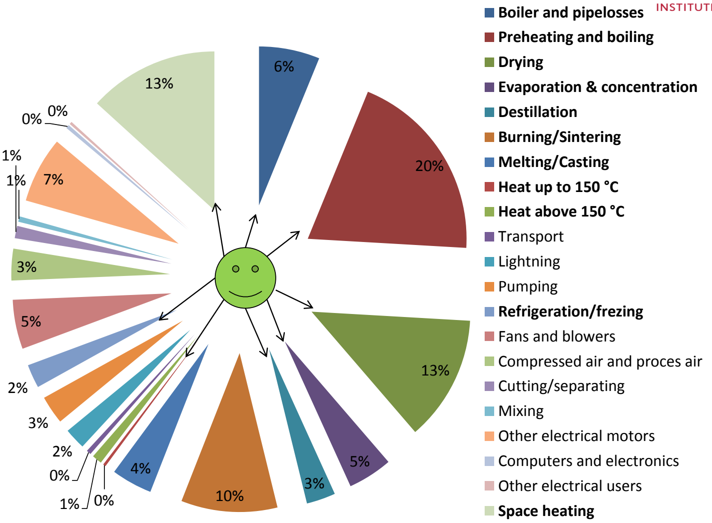

Source: Kortlægning af erhvervslivets energiforbrug, November 2008, Energistyrelsen

Task 1: Market overview, Barriers

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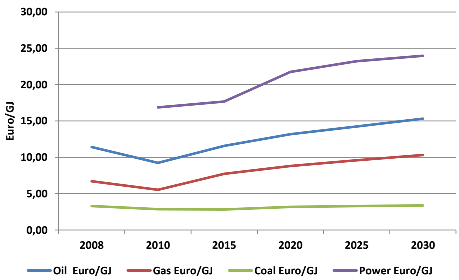
- **Profitability**; Minimum COP at 2,3 before positive return compared to gas.
- **Focus on reutilization for space heating;** Danish energy tax system is a big challenge.
- **Knowledge:** Lack of knowledge and experience also constitutes a challenge in the industry.
- **Focus on product not energy:** The manufacturers are having focus on their product, the energy is not their main focus.



Source		2008	2010	2015	2020	2025	2030
Oil	Euro/GJ	11,42	9,25	11,57	13,18	14,24	15,32
Gas	Euro/GJ	6,71	5,54	7,73	8,82	9,59	10,33
Coal	Euro/GJ	3,29	2,87	2,82	3,19	3,30	3,38
Power	Euro/GJ		16,86	17,68	21,76	23,22	23,96
Power/gas ratio		0,00	3,04	2,29	2,47	2,42	2,32

Source: IEA fuel predictions ,World Energy Outlook 2008

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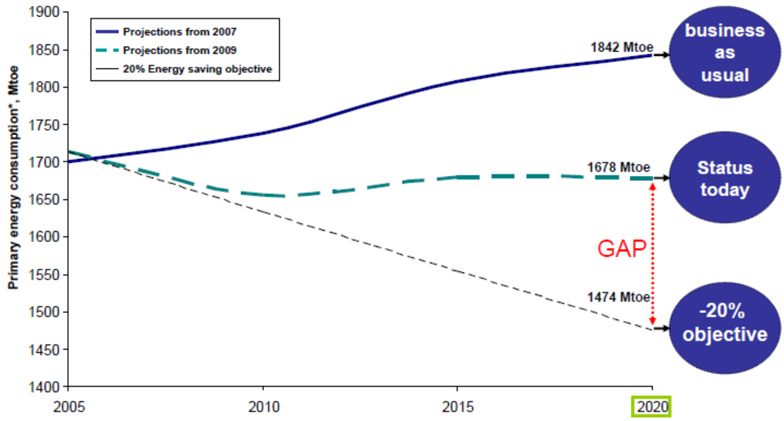
Task 1: Market overview, Drivers



Source: European commission

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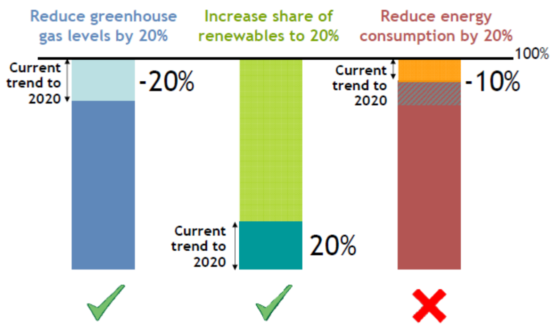
Task 1: Market overview, Drivers

Political:

- **EU : Energy Efficiency Directive will lead to:**
  - **Waste heat recovery (CHP) obligation for new and existing power & industrial plants**
  - **Network tariffs design to encourage offering services for consumers allowing them to save energy & control consumption**
  - **Member States to create incentives for SMEs and large companies to undergo energy audits.**

MEETING ALL THREE "20-20-20 BY 2020" GOALS BECOMES A MATTER OF URGENCY



Task 2: Modeling calculation and economic models

- Make SWOT analyses of available software and calculation procedures for application for different sectors.
- Analyze and update of existing models from Annex 21, where does the heat pump fit and how does it fit.
- Use the analyses of tools and findings of task 1 to determine the gaps, needs and possibilities for new model development.
- Examine the possibilities to make software available.

Software Tools

- Tools for process integration
- Tools for balancing and flow sheeting simulation
- Tools for general purpose optimization and process synthesis
- Novel Tools
- Future trends

Recovery sources and Heating loads

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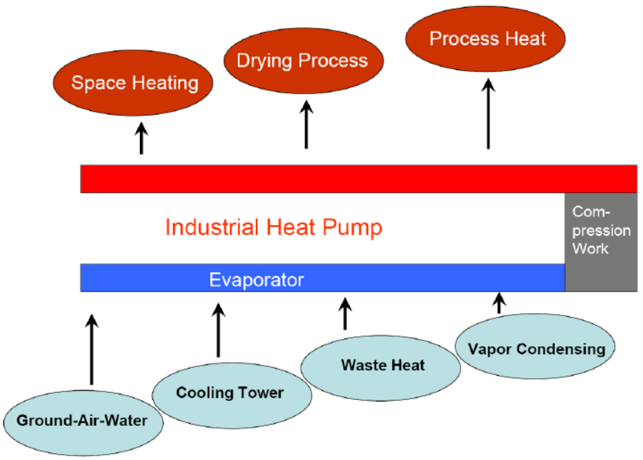

Task 2: Modeling calculation and economic models

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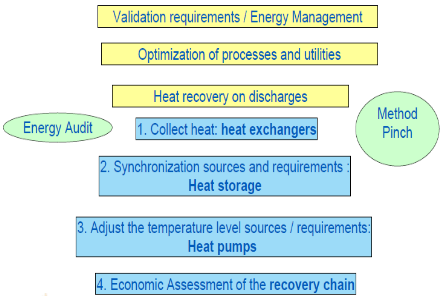
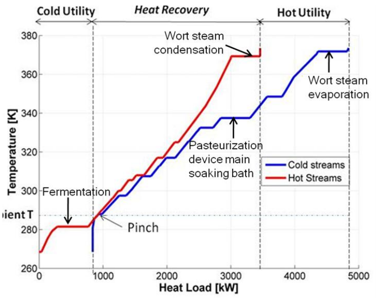
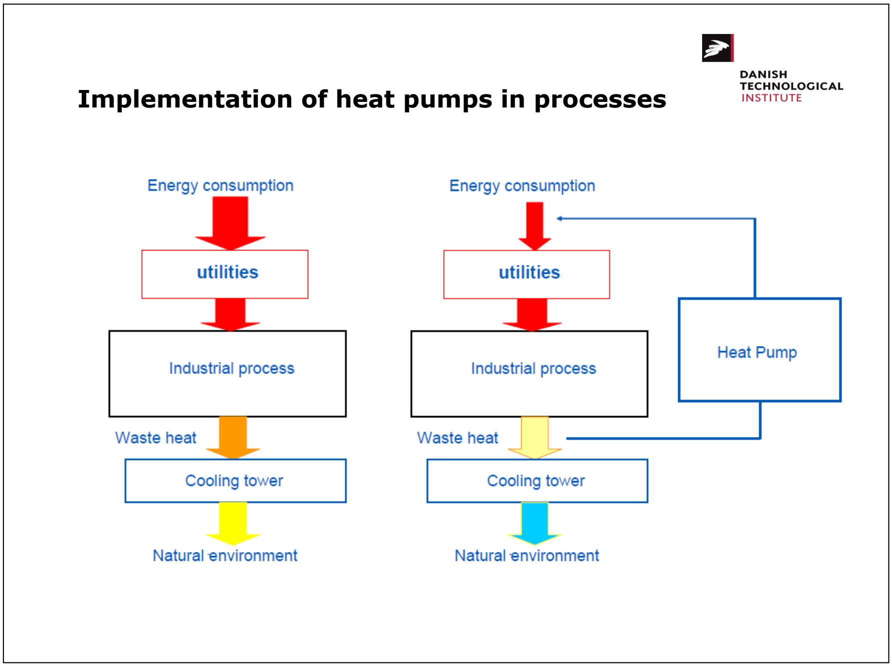
- Map energy demands/flows (loads of heating & cooling) in the processes
- Map operation times and time of day/week
- Calculate average demands
- Map operation times and time of week
- Identify heat recovery possibilities at "average" demands (Pinch analysis)
- Check the opportunity for heat recovery taking time into account
- Check the opportunity for heat recovery using storages
- Check the possibility to implement heat pump in the production
- Check the physical possibilities taking place and distances into account.






Implementation of heat pumps in processes
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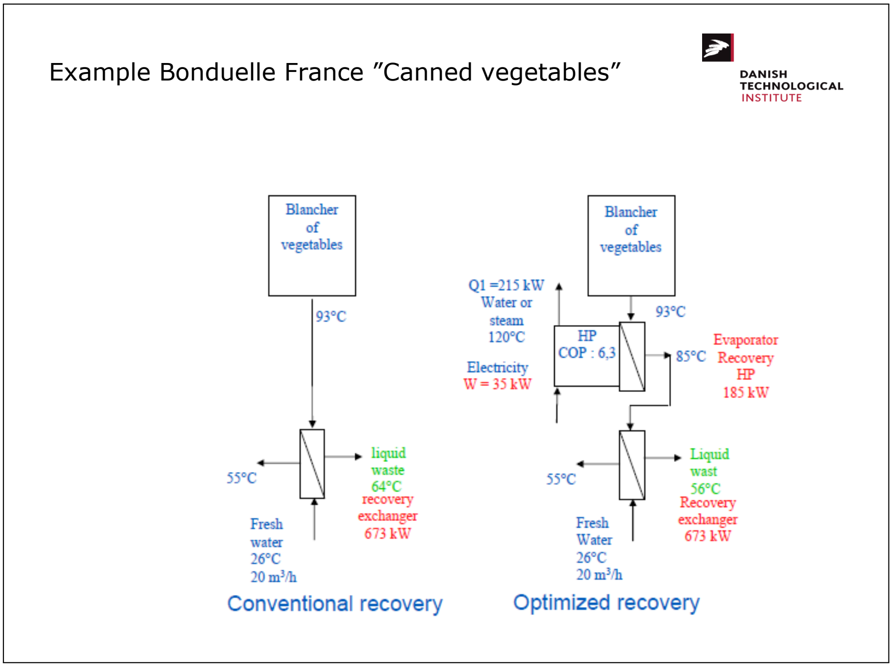
Energy consumption
utilities
Industrial process
Waste heat
Cooling tower
Natural environment
Energy consumption
utilities
Industrial process
Heat Pump
Waste heat
Cooling tower
Natural environment


Example Bonduelle France ”Canned vegetables”
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Blancher of vegetables
93°C
55°C
liquid waste 64°C
recovery exchanger 673 kW
Fresh water 26°C 20 m³/h
Conventional recovery
Blancher of vegetables
Q1 =215 kW Water or steam 120°C
93°C
HP COP : 6,3
85°C
Evaporator Recovery HP 185 kW
Electricity W = 35 kW
55°C
Liquid wast 56°C
Recovery exchanger 673 kW
Fresh Water 26°C 20 m³/h
Optimized recovery

Task 3: Technology: High temperature heat pumps, process technological integration, refrigerants

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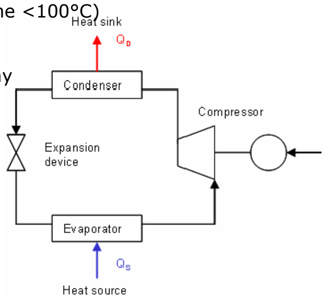
Heat pump closed compression cycles
Processes, different characteristics, different applications?

- Traditional refrigeration cycles (NH3) <100°C
  - Cold side: Evaporation: No glide
  - Hot side: Condensation: No glide
  - Researc/Projects going on in : Norway, Denmark, Sweden
  - Companies: JCI, GEA, Thermea and more

- Traditional refrigeration cycles (HC Propane/Isubutane <100°C)
  - Cold side: Evaporation: No glide
  - Hot side: Condensation: No glide
  - Researc/Projects going on in : Denmark, Germany
  - Companies: JCI, Bundgaard, and more

- Traditional refrigeration cycles (R245fa <140°C)
  - Cold side: Evaporation: No glide
  - Hot side: Condensation: No glide
  - Researc/Projects going on in : France
  - Companies: EDF, JCI France, CIAT



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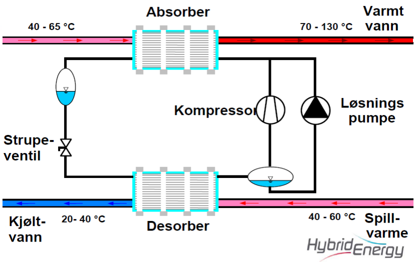
Technology: High temperature heat pumps, process technological integration, refrigerants

Heat pump closed compression cycles
Processes, different characteristics, different applications?

- Transcritical CO2 <130°C
  - Cold side: Evaporation: No glide
  - Hot side: Gas cooling: Glide
  - Researc/Projects going on in : Norway, Denmark, Germany, Italy
  - Companies: Advansor, Thermea,

Absorption-compression hybrid

- Hybrid (H20 / NH3) <250°C
  - Cold side: Evaporation: Glide
  - Hot side: Condensation: Glide
  - Researc/Projects going on in : Norway, Denmark, Korea
  - Companies: Hybrid, IM



Technology: High temperature heat pumps, process technological integration, refrigerants

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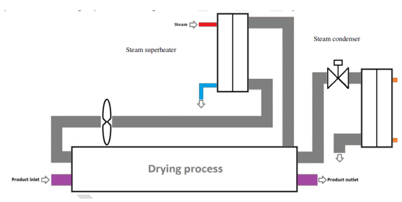
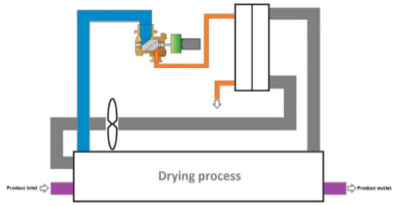
Mechanical vapour recompression (MVR)

- Water Vapour <250°C
  - Cold side: Evaporation: No glide
  - Hot side: Condensation: No glide
  - Researc/Projects going on in : Denmark, Japan, Germany
  - Companies: DTI, JCI, Kobelco





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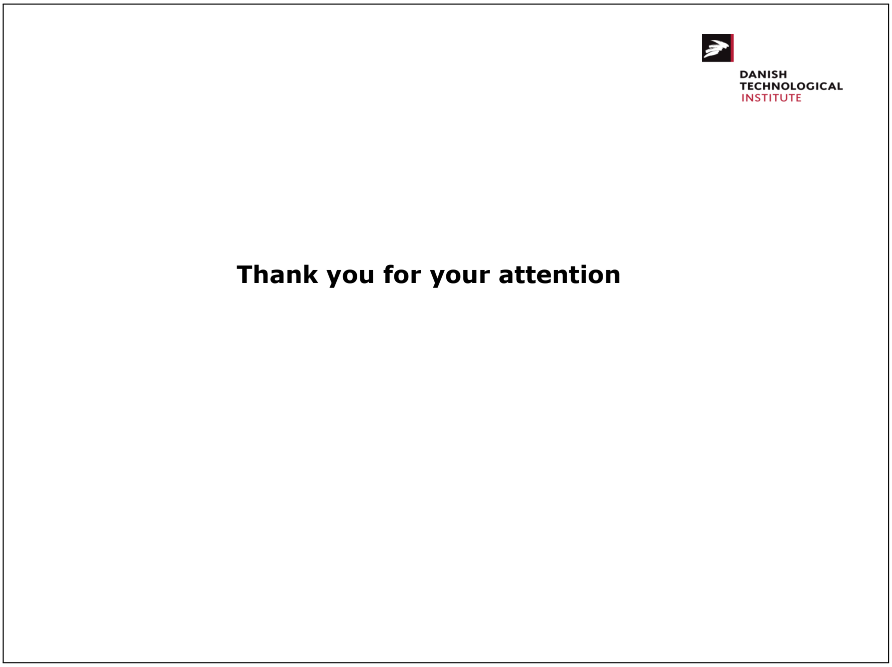
Conclusion

- There is a **large potential** for implementing industrial heat pumps
- The main **barriers are economical**, and fossile fuel prices, tax structure.
- The **drivers are political** driven the 20-20-20 targets.
- Structured energy audits are essential for a good implementation
- Heat exchange before implementation of heat pumps
- Technology are available
- The right right technology, depends on the proces.



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3.10 Refrigerant Flow in Vertical Pipes

Lars Reinholdt *(lre@teknologisk.dk)*
DTI

Timetable ▲
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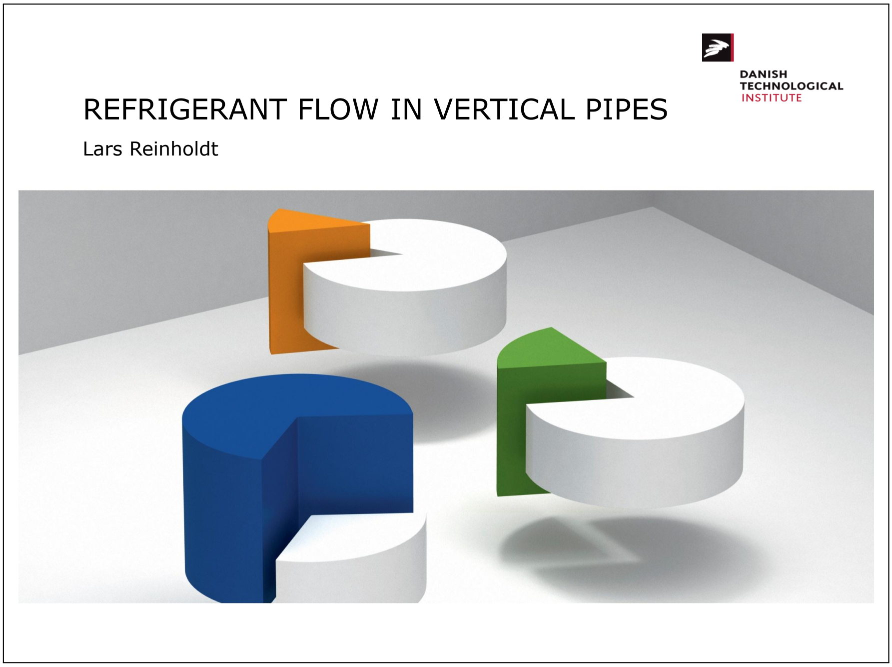

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Content

- Why two phase flow in vertical piping?
- Theory
- Experiments
- Challenges in designing "riser" systems
- Other solutions
- Conclusions

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Why two phase flow in vertical piping?

Flooded evaporators!!!

Why not DX:

- Higher evaporating temperature: Traditional DX evaporators need superheating of the return gas resulting in a lower evaporating temperature
- Ammonia is quite difficult to control in DX operation
- Higher heat transfer coefficient more even temperature distribution

Challenge:

- Liquid over feed results in wet discharge gas form the evaporator
- Traditionally liquid and gas is separated in the plant room: The liquid overfeed has to be returned along with the return gas
- Result: Wet return lines

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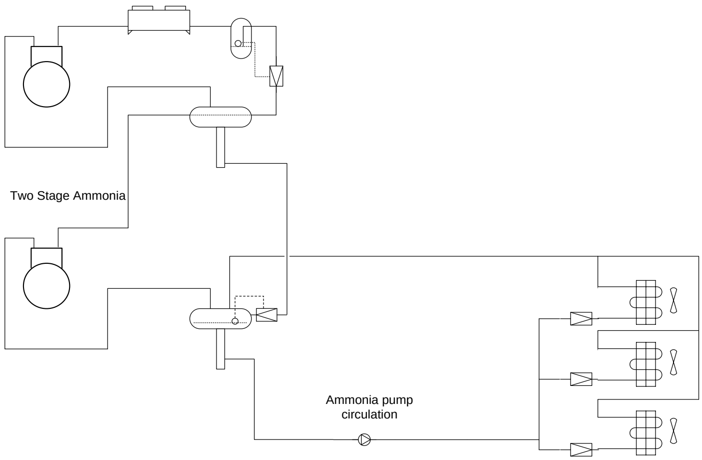
Typical industrial system design:



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Wet return line

- As long as the wet return line is lower and inclined in the direction of the separator the design is not that difficult
- This solution is in far most cases not practical
- Normally the return line is on the roof having the evaporators several metes below…
- We need a vertical piping serving the transportation of two phase mix of liquid and vapor: The RISER

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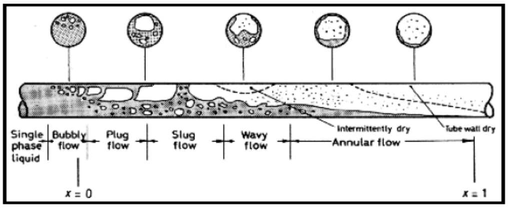
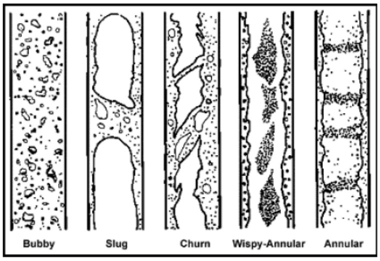
Theory

Flow patterns

- Horizontal piping



- Vertical piping



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Theory

- General formulation

$$\Delta P = \Delta P_{grav} + \Delta P_{fric} + \Delta P_{acce}$$

- Redusing to

$$\Delta P = \Delta P_{grav} + \Delta P_{fric}$$

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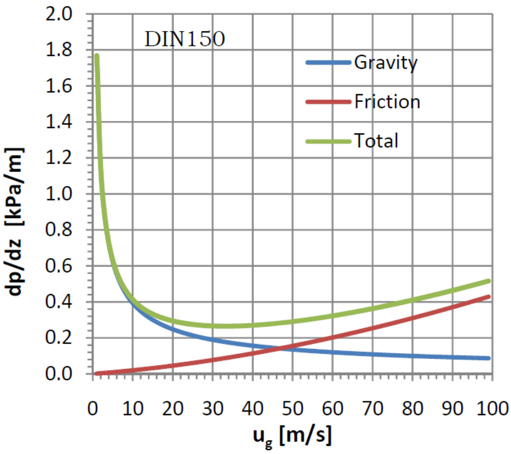
Theory

Keeping the pipe size the same:

- At low gas flow: Gravitional pressure drop dominating
- At high gas flow: Frictional pressure drop dominating



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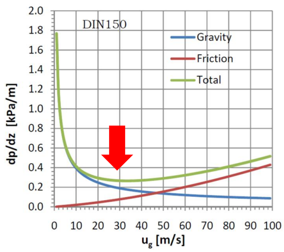
Modelling

Two phase flow

- Model for the void fraction
- Model for the pressure drop

From a design point of view:

- Identification of minimum pressure drop: Flow reversal



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Flow reversal

Two models

- Wallis (air/water in small tubes)
- Pushkin and Sorokin (6 to 399 mm tubes)

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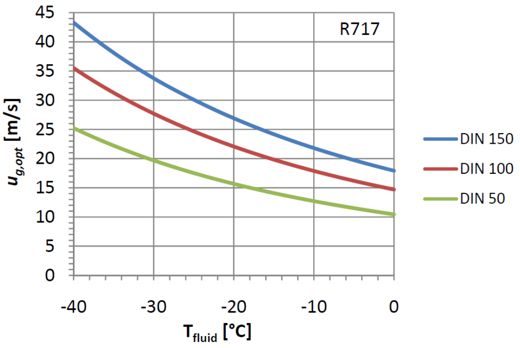
Flow reversal

Wallis

$$u_g^* = u_{g,rev} \sqrt{\frac{\rho_g}{g\ d\ (\rho_l - \rho_g)}} \approx 1$$

- Flow reversal at u_g^* = 1.06 to 1.12



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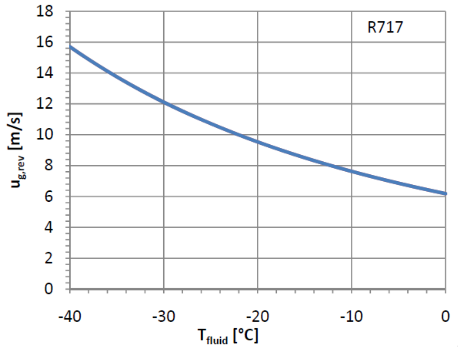
Flow reversal

Pushkin and Sorokin

- Found flow reversal at Kutatelatze no Ku = 3.2

$$Ku = u_{g,rev}\,\rho_g^{0.5}\,\left[g\sigma(\rho_l - \rho_g)\right]^{-0.25} = 3.2$$



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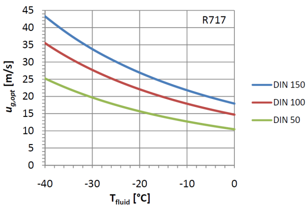
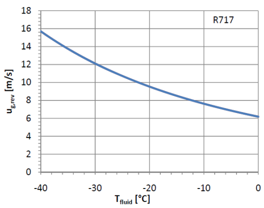
Flow reversal

The two models: Large difference depending on pipe size





- Wallis and Kuo developed:

$$D^* = d\left[\frac{g\,(\rho_l - \rho_g)}{\sigma}\right]^{\frac{1}{2}}$$

- …and suggest

  Wallis at D* < 10.24 and Pushkin and Sorokin at D* >= 10.24
  For DN 50 (2") : D* ~ 25

Models based on air / water experiments…

- Valid for Ammonia ????

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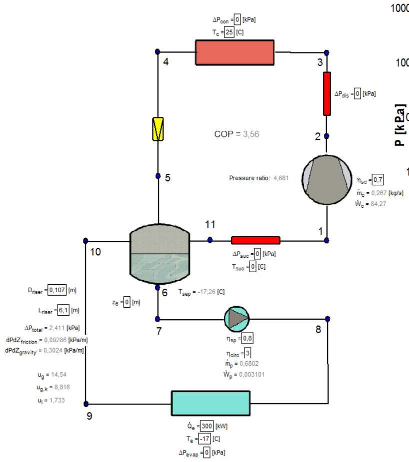
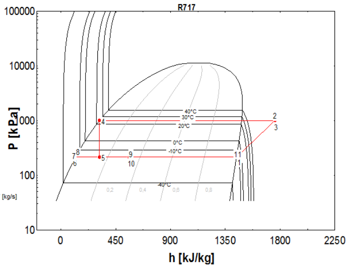
Does the pressure drop matter?





	h [kJ/kg]	P [kPa]	T [°C]	s [kJ/kgK]	x
1	1441	214,4	-17,26	5,862	
2	1757	1003	130,1		
3	1757	1003	130,1		
4	317,5	1003	25		
5	317,5	214,4	-17,26		0,149
6	121	214,4	-17,26		
7	121	214,4	-17,26		
8	121	216,8	-17,26		
9	562	216,8	-17		0,3333
10	562	214,4	-17,26		0,334
11	1441	214,4	-17,26		

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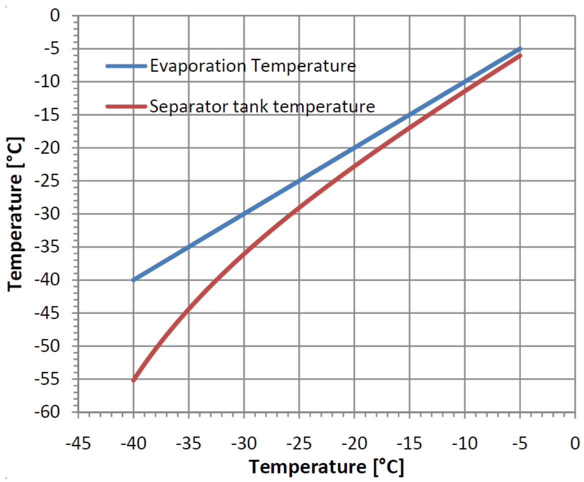
Pressure drop



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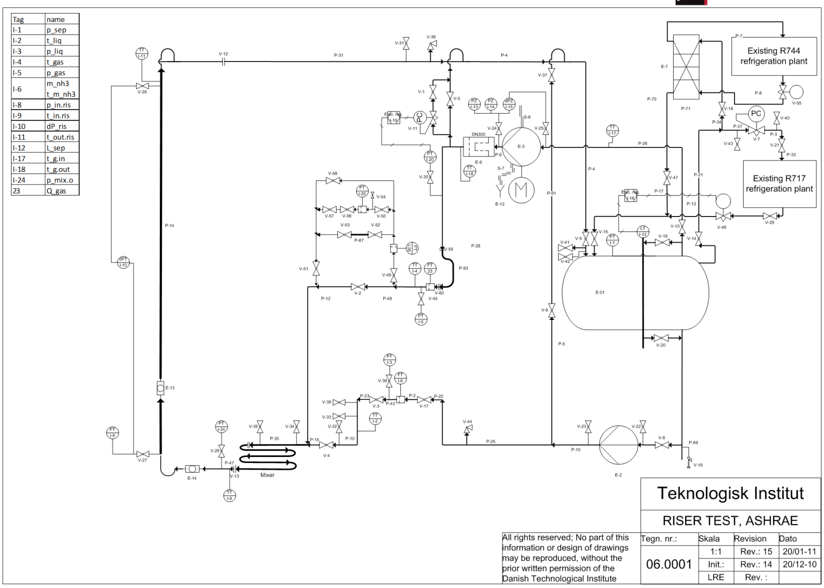
Experiments

ASHRAE funded research work (RP 1327)

Identify flow reversal and measure pressure drop:

- Pipe size: 50mm (2”), 100 mm (4”) and 150 mm (6”)
- Temperatures: -40°C (-40°F), -29°C (-20°F), -18°C (0°F) and -6°C (20°F)
- Circulation rates: 2, 3, 4, 5 and 10

Test rig



Overview (old!!!!!)

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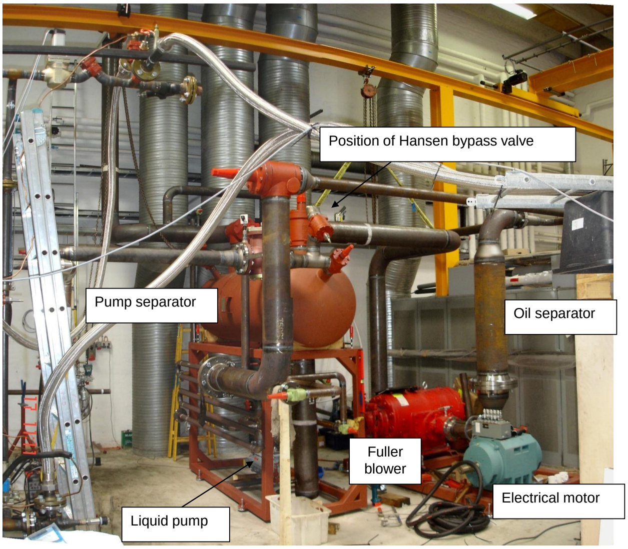

Liquid flow measuring system

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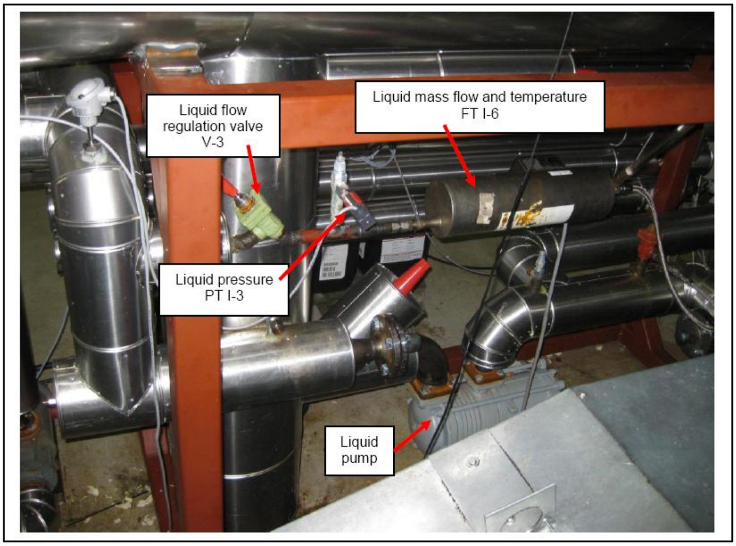

Gas flow measuring system

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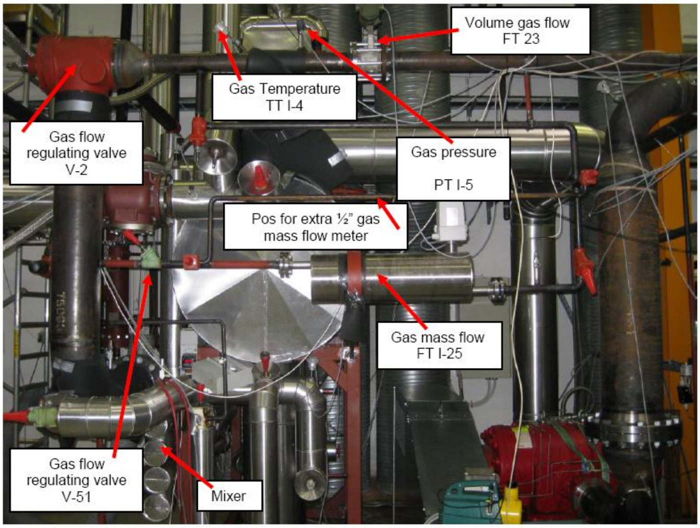

Gas flow measuring system incl. orifice

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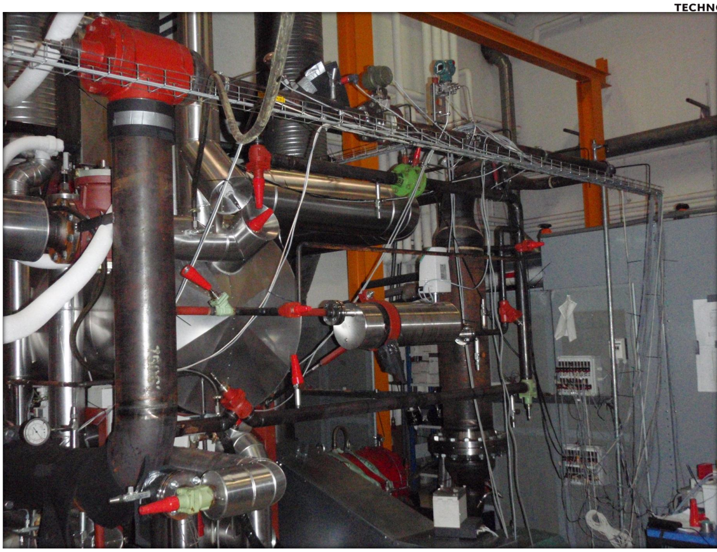

Riser

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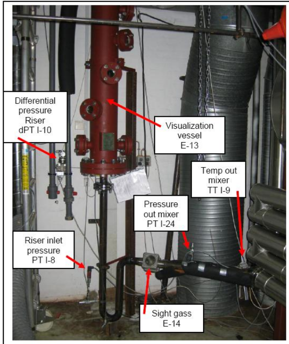

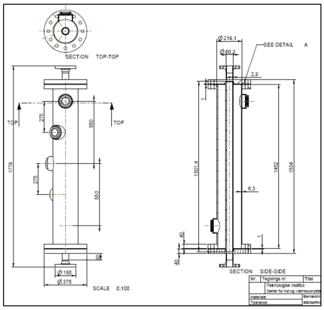

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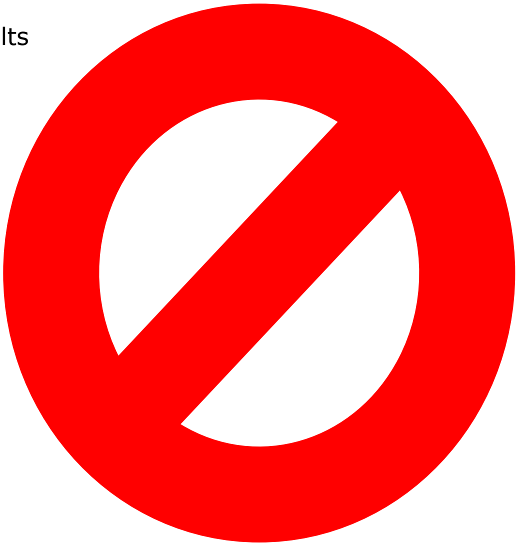
Results



Visualizations

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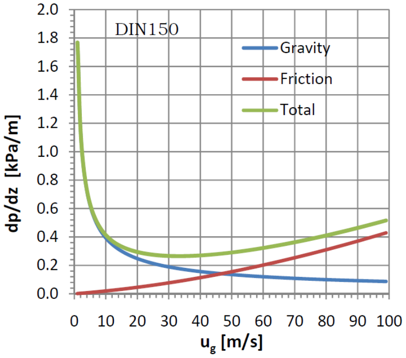
Challenges in designing "riser" systems

- Part load
- Change in load
- Sudden change in pressure



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Other solutions

- Why the riser???
- Due to the evaporator not the system
- Can we make a dry return and still overfeeding the evaporator?
- YES!

Low Pressure Receiver (LPR)

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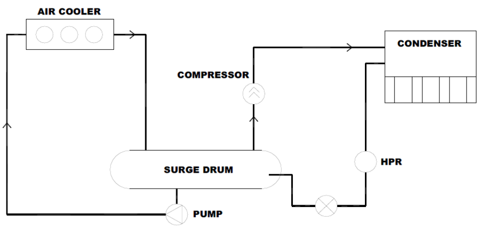
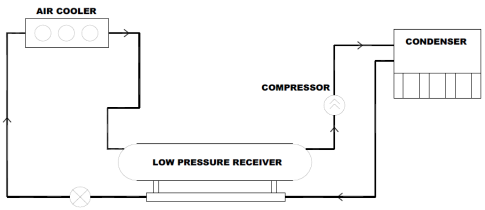
- Traditional design



- LPR



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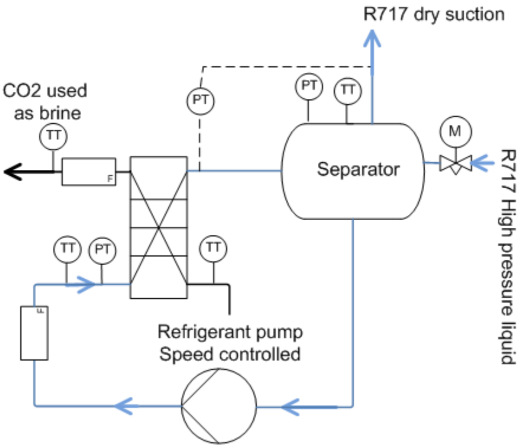
Local separation



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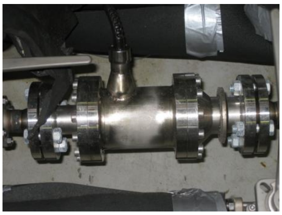
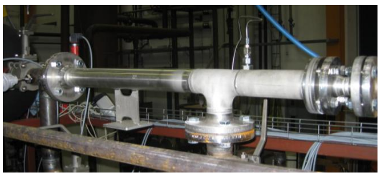
Local separation

- Self circulating



- Pump



- Ejector

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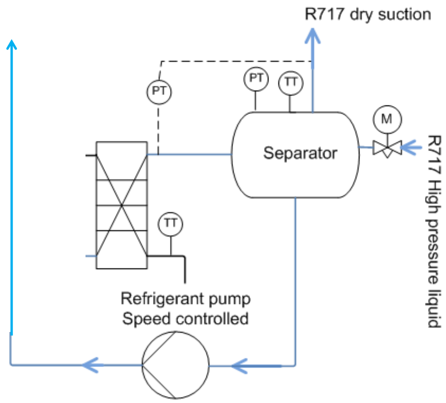
Local separation

- Dry return + pumped return of liquid



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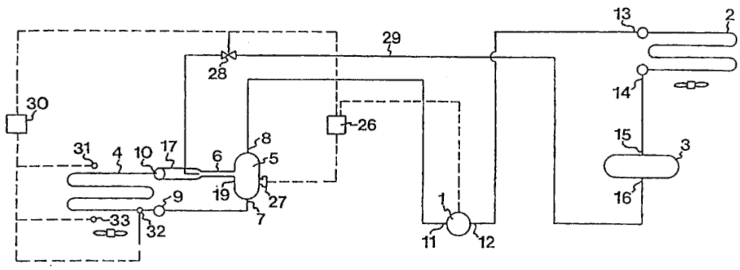
Frigoscandia LVS



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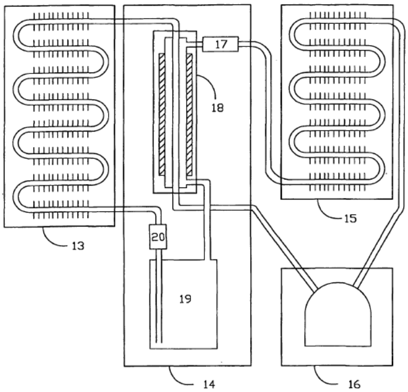
Selfregulating system by L. Zimmermann

- Compressor (16)
- Condenser (15)
- Heat sensitive valve (17)
- Receiver (19)
- Pressure sensitive valve (20)
- Evaporator (13)
- Heat exchanger (18)



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Conclusion

- Flooding of evaporators are often beneficial
- In traditionally designed (industrial) systems wet return is widely used and risers are needed
- The design is not strait forward as flow reversal can happen
- A minimum pressure drop is required in order to have a stable liquid return
- Pressure drops has an impact on the system efficiency especially at low evaporating temperatures
- The correlations / mathmatical models used for design is being verified
- Other system design having both flooded evaporators and dry return is available

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3.11 Cooling Towers of the Future

Peter Schneider *(psc@teknologisk.dk)*
DTI

Timetable ▲
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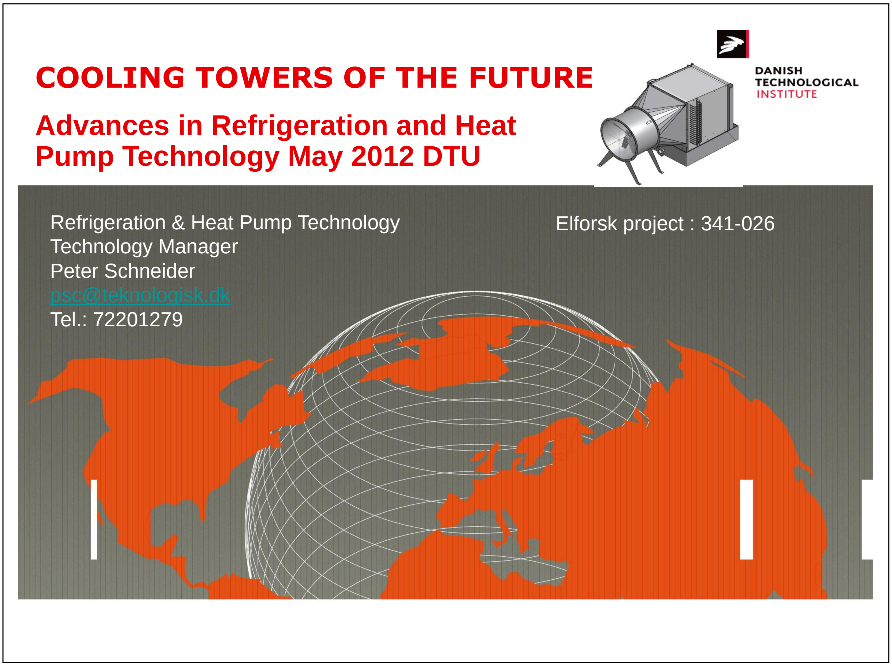

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Nickname of the prototype: Rhino!

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Outline

- Cooling tower theory
- Objectives of the project
- Presentation of the project participants
- Design of the cooling tower
- Testrig and measurements
- Calculation program
- Energy savings

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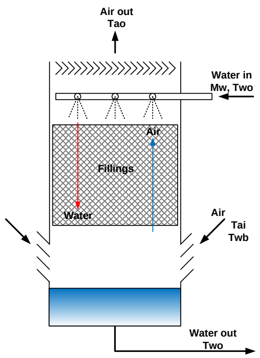
What is a cooling tower?



A cooling tower is device which rejects waste heat to the atmosphere by cooling hot water to a lower temperature.

The type of heat rejection in a cooling tower is termed "evaporative". A small portion of the water is evaporated into the moving air stream to provide cooling to the rest of that water stream.

The heat from the water stream transferred to the air stream increases the temperature of the air and its relative humidity to 100% and that air is discharged to the atmosphere.

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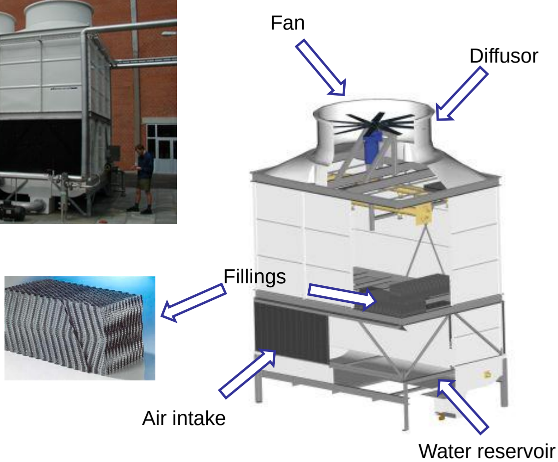
Existing cooling towers



Cooling tower theory

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The heat transferred from the hot water to the cold air is a combination of mass and heat transfer. The two can be combined by using enthalpy as driving potential between the water and air (Water as saturated air (r=1) on the surface of the fillings).

Equation for heat transferred in a cooling tower

$$dQ=K*dA*(h_{water}-h_{air})$$

K : Mass transfer coefficient (U-value in a HX)
A : Heat transfer area

$h_{water}-h_{air}$: Enthalpy difference between water and air (DT in a HX)

Heat balance : $Q=L_w*C_{pw}*(T_{wi}-T_{wo})=G_{air}*(h_{airo}-h_{airi})$

eller

$h_{airo}=h_{airi}+L/G*C_{pw}*(T_{wi}-T_{wo})$, Operation line

Enthalpy

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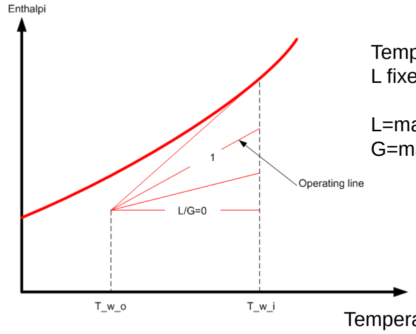

Temperatures: Twi; Two and Twb fixed
L fixed

L=mass flow water
G=mass flow air

- The slope of the operating line is L/G
- The driving potential dH (DT i a HX) changes with the L/G ratio

Merkel number, named after Dr. Merkel

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$$Me = \int \frac{Cpw \cdot DT_w}{h_w - h_{air}} = \frac{K \cdot A}{L}$$

Left side of equation is dependant on Twi, Two, Twb and LG

Right side of equation is dependant on the fillings : Mass transfer coefficient and the L/G ratio

The characteristic of the cooling tower tower filling is described as:

$$Me = C \cdot \left(\frac{L}{G}\right)^m$$

C and M are constants for the cooling tower filling in question.

Typical values: M=-0.6 to -0.8
C= 1 to 3

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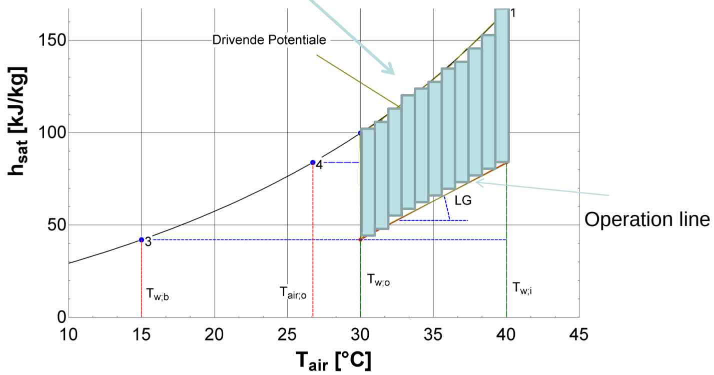
The driving potential DH changes through the tower and the overall DH is calculated by integration in small steps through the tower

$$Me = \int_{Twi}^{Twu} \frac{Cw \cdot DTw}{hw - hl}$$



Easier way to calculate the integral

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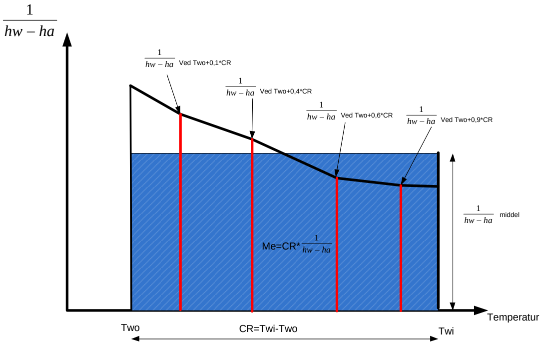

No integration – straight forward calculation, deviation < 1% compared to numerical integration. Chebyschew polynomia

$$\frac{KaV}{L} = Cw\int_{tw1}^{tw2} \frac{dtw}{hw-ha} = (tw_2 - tw_1) \times [(1/Dh_1) + (1/Dh_2) + (1/Dh_3) + (1/Dh_4)]/4$$

where, Dh_1 = value of (hw - ha) at a temperature of CWT + 0.1 x Range
Dh_2 = value of (hw - ha) at a temperature of CWT + 0.4 x Range
Dh_3 = value of (hw - ha) at a temperature of CWT + 0.6 x Range
Dh_4 = value of (hw - ha) at a temperature of CWT + 0.9 x Range

Cooling tower fillings and capacity curves

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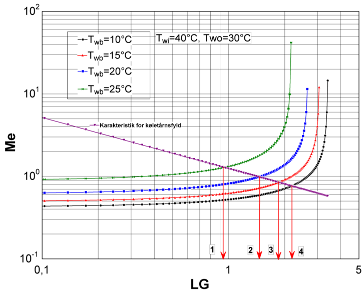
The following equation describes the cooling tower filling:

$$Me = C \cdot \left(\frac{L}{G}\right)^M$$

C and M are constants for the cooling tower filling in question.

Typical values: M=-0.6 to -0.8
C= 1 to 3



Why is a cooling tower energy efficient ?

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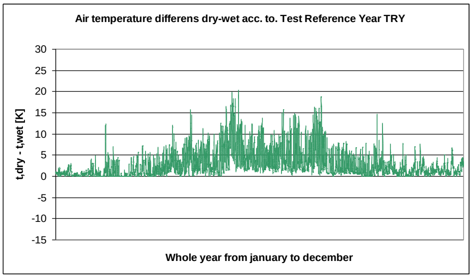

- The dry temperature is higher than the wet temperature
- A dry cooler sees the dry temperature
- A cooling tower sees the wet temperature

- Lower condensing temperature for a refrigeration plant with cooling tower => energy savings (more about that later)

Objectives of the project

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- To develop a cooling tower that is safe with regard to health and that uses rainwater as make-up water
- To substitute dry cooler with cooling towers
- To develop a competitive cooling tower
  - In terms of price and performance
- To prepare a design programme for the new cooling tower
- To prepare a calculation programme in order to select a dry cooler or a cooling tower and calculate energy savings

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Project participants

- Danish Technological Institute
  Refrigeration and Heat Pump Technology, Project manager
- Vestas Aircoil A/S – Producer of cooling towers
- MultiWing A/S – Fan producer
- Silhorko A/S – Water treatment
- Nyrup Plast A/S – Rainwater tanks
- Rambøll A/S – Calculation program
- Municipality of Skive – End-user

Project funded by Elforsk project no. 341-026
2009-2011

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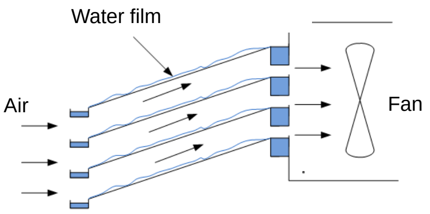
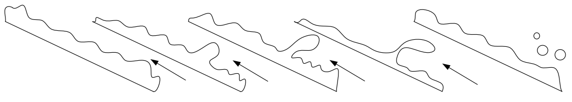
The principles of the new cooling tower



The concept has showed that no droplets are carried with the air



If the water film is turbulent, waves are created and it is possible that droplets can be created

Studies of the air side, CFD

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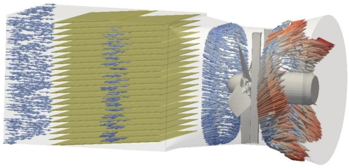
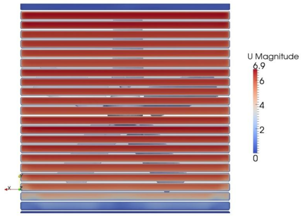
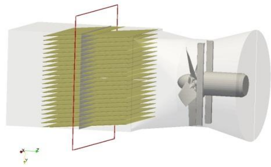


Dimensions of plates:
- Width: 2 m
- Length: 2 m
- Height: 2 m

Number of plates: 20
Slope of plate: 5°
Air inlet velocity: 5 m/s
Fan rotations: 400 rpm





Calculated pressure drop

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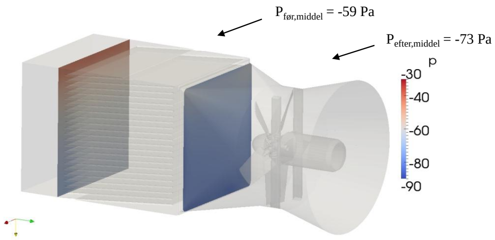

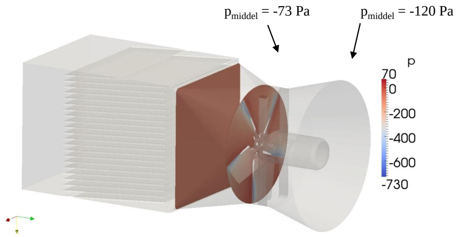

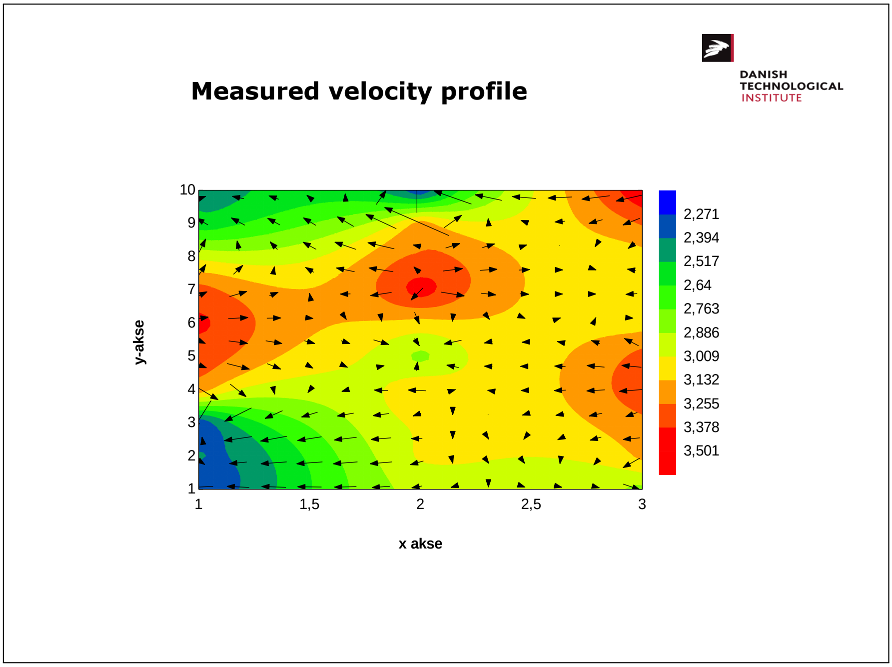
Measured velocity profile
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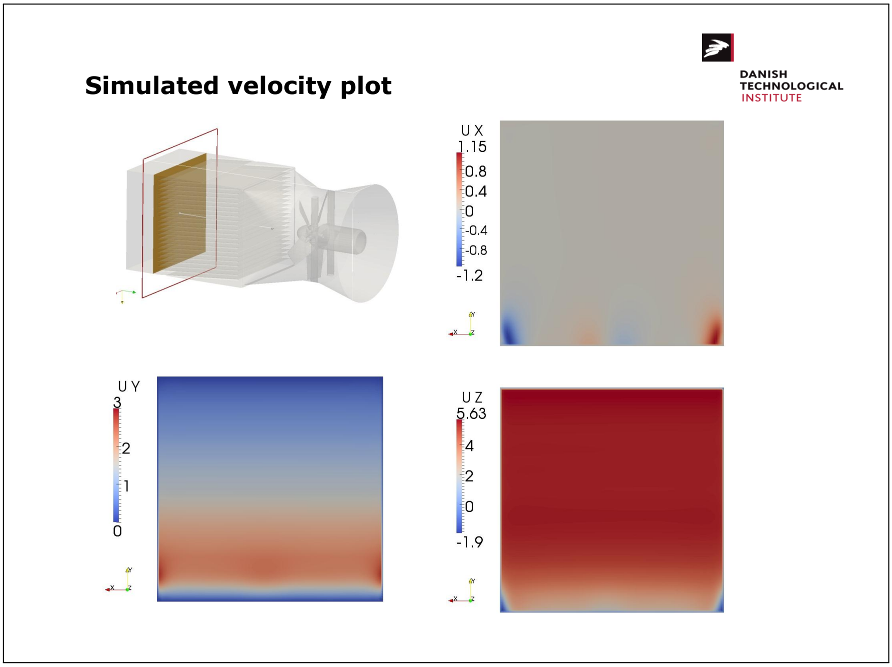
y-akse
x akse
10
9
8
7
6
5
4
3
2
1
1
1,5
2
2,5
3
2,271
2,394
2,517
2,64
2,763
2,886
3,009
3,132
3,255
3,378
3,501


Simulated velocity plot
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U X
1.15
0.8
0.4
0
-0.4
-0.8
-1.2
U Y
3
2
1
0
U Z
5.63
4
2
0
-1.9

Simulation of airflow across trays

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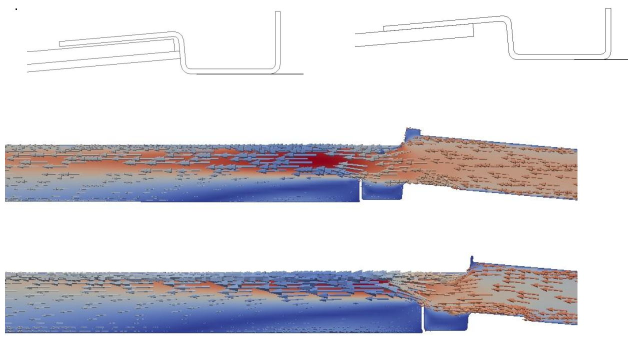

Design audit 1: Δp = 18 Pa
Design audit 2: Δp = 35 Pa

Discharge trays

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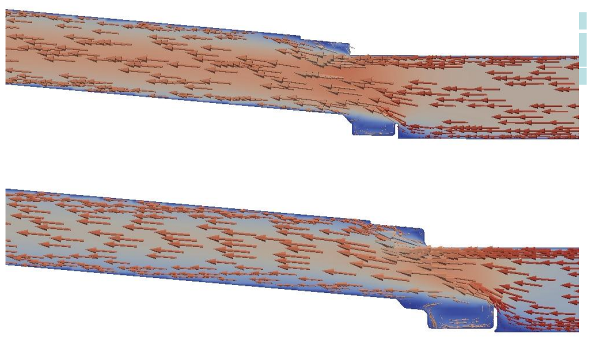

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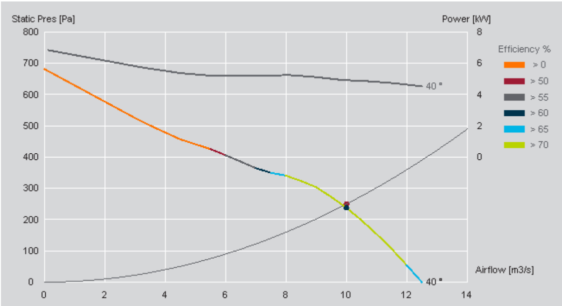
Choice of fan



On the basis of CFD calculations, a fan from MultiWing was chosen with an airflow of 10 m^3/s and a static drop of pressure across the plates of 250 Pa.

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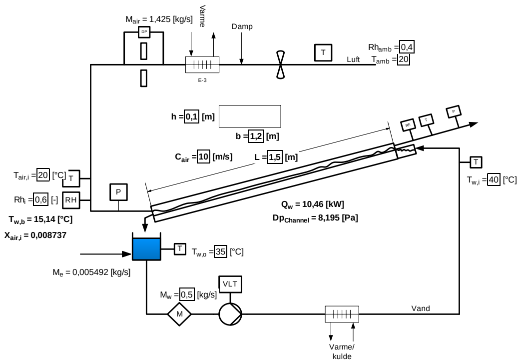
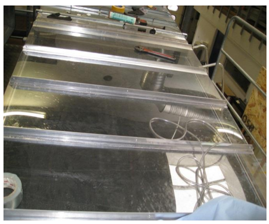
Measurement setup for determination of plate characteristic





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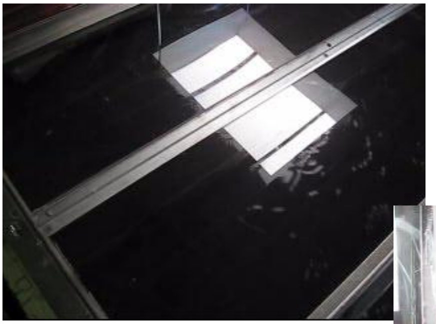
Films about water



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Test Matrix

Water temperture T_{wi} [°C]	Air veleocity [m/s]	Masseflow water/plate [kg/s]
30	1	0,05
35	2	0,10
40	3	0,15
	4	
	5	

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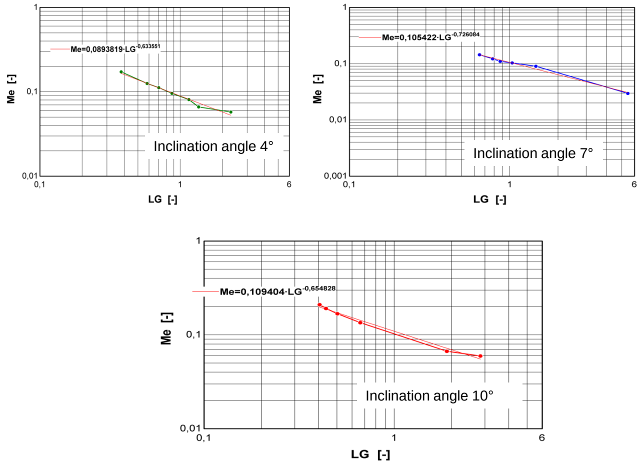
Plate characteristic at 3 angles of inclination



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Distribution of water on plates

Not included in the presentation as it has not been implemented on the cooling tower, but it shows that it is possible to make a plate that actively gives good distribution of the water.

In the long run, the below surfaces could be used for good liquid distribution.

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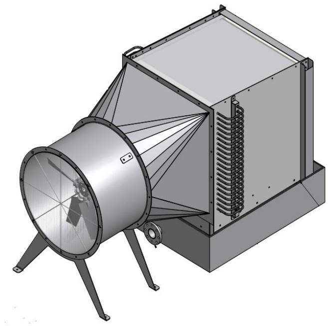
The design basis had been determined and Vestas Aircoil started building the prototype.



And it wasn't easy! Water distribution – tolerances – plastic plates – stainless steel plates etc.

Pictures from the production of the prototype

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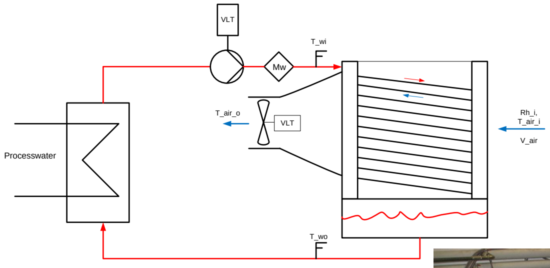
Measurement setup



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Measurement programme

Water temperature T_{wi} [°C]	Air velocity [m/s]	Mass flow per plate [kg/s]
30	1	0.05
35	2	0.10
40	3	0.15
	4	
	5	

Pictures of the prototype Cooling Tower

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Plate characteristic and performance

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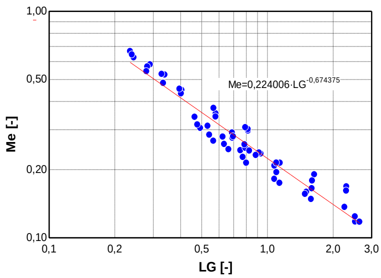

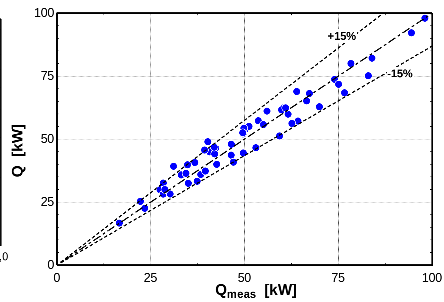

Correlation for the Merkel number

$Me = 0{,}22*LG^{-0{,}67}$

Comparison between correlated and measured capacity . Deviation <15 %

Capacity measured on the waterside Q=m*Cp*DT

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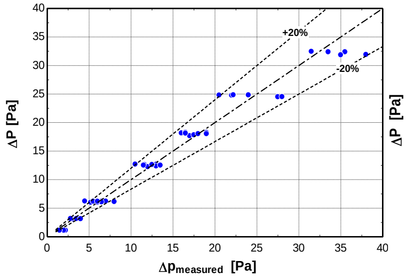
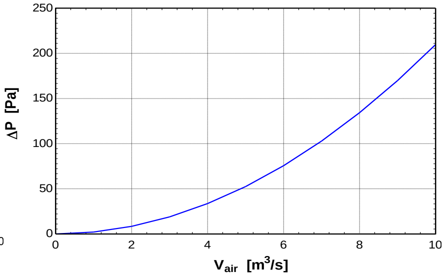
Drop of pressure





Vair=Cair, da Aface=1m2

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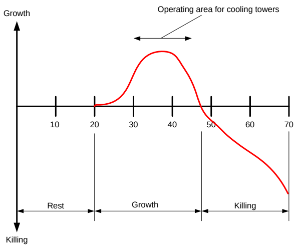
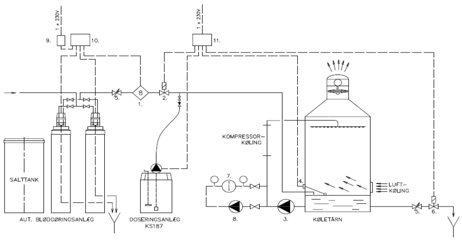
Water treatment and bacteria (legionella) growth



Minimization
- of bacteria growth
- scaling
- corrosion
- Biofilm
- Cooling towers forbidden in large cities

Through softening, the calcium and magnesium salts in the water are exchanged for sodium salts which do not cause the disadvantages of hard water.

By using rainwater (soft water), there is not much need for softening.



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Legionella problems

- Spread from cooling towers from airborne aerosols (d = 1-5 micro meter)
- The growth of legionella is in the temperature area, where cooling towers operate
- The legionella bacteria “eats” the nutrients, which are washed out of the air and are accumulated in the water reservoir of the cooling tower
- The water of the cooling tower has to be replaced to avoid the problems above
- The cooling tower concept has proven that no aerosols are created for a specific range of water and air flows

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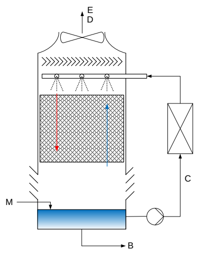
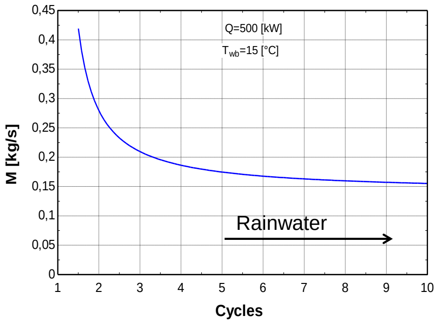
Bleed off





Cycles = xc/xm

M= Make up water, B= Bleed off, E= Evaporated, D=Drip loss, C= water flow

Because rainwater is soft (no minerals dissolved) the number of cycles is higher than when normal water is used.

Collection and accumulation of rainwater

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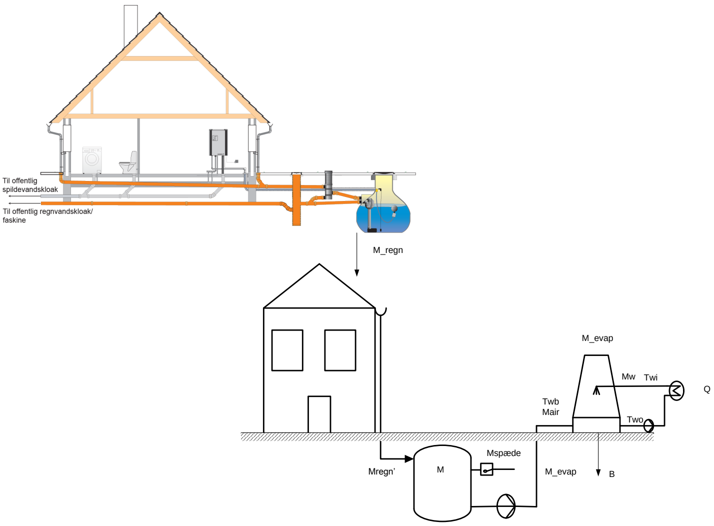

Calculation programme (LCC, energy, dry cooler, cooling tower)

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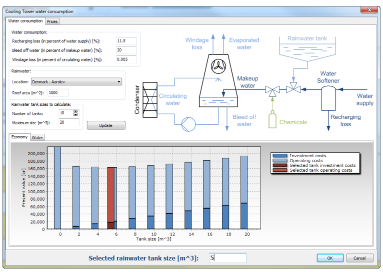
The yearly energy consumption for different types of refrigeration systems can be calculated in PackCalc II.

In the course of the project, cooling tower models have been implemented in PackCalc II so the yearly energy consumption of a refrigeration system with cooling tower as heat drain can be compared with other heat drain systems (e.g.: Dry cooler).

The cooling tower models have been prepared by Vestas Aircoil A/S, Rambøll and Danish Technological Institute.

It is possible to download Pack Calculation II from the following homepage:

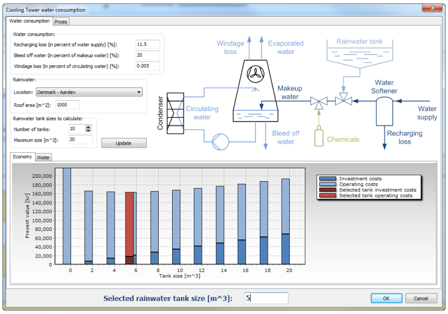
http://www.ipu.dk/IPU-Teknologiudvikling/Koele--og-energiteknik/Downloads/PackCalculation.aspx



Load profiles, input, refrigeration system, geography

In the programme the water consumption consists of:

- ”recharging loss” is the amount of water that is used to regenerate the water softener
- Bleed off
- Drift loss, the amount that is lost due to splashes.



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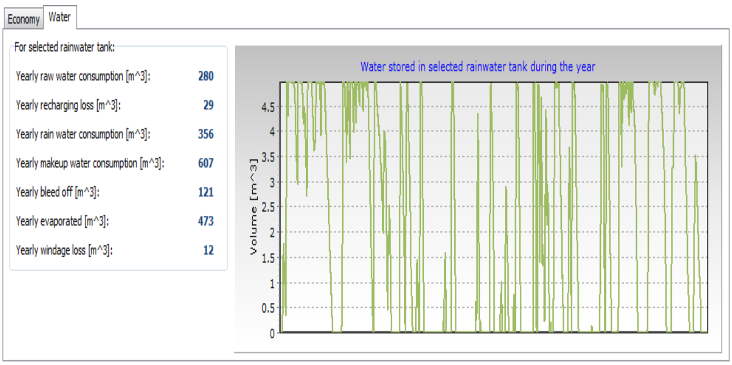
Rainy day profiles



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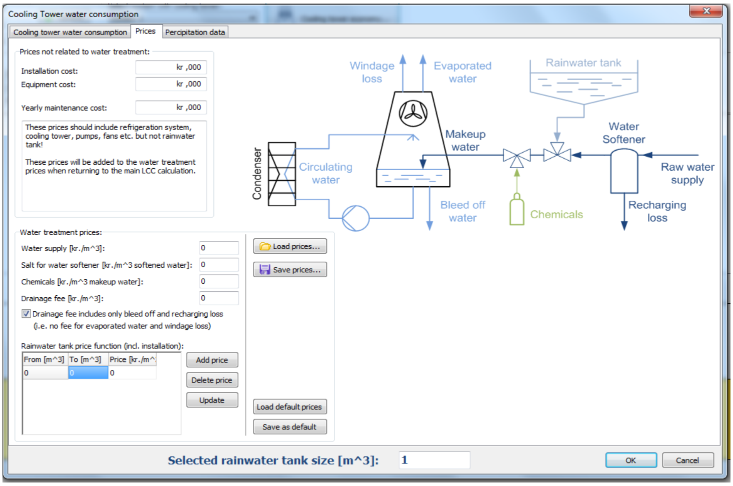

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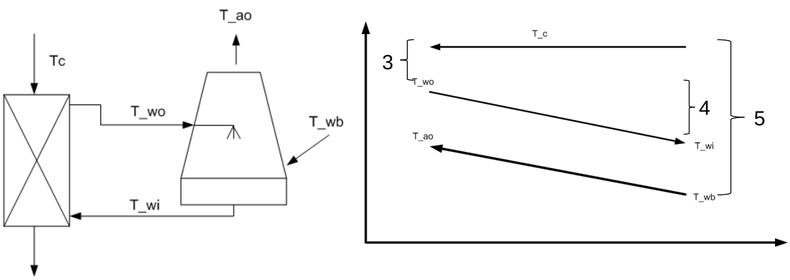
Temperature definitions in PackCalc, cooling tower



Standard preferences
W_fan+W_pump=1.5%*Q_c
Example :

Q_c=500 kW => W_fan+W_pump = 7.5 kW
15% to pump = W_pump=0.15*7.5 =1.1 kW
W_fan = The rest to the fan = 7.5-1.1 =6.4 kW

Default values, cooling tower

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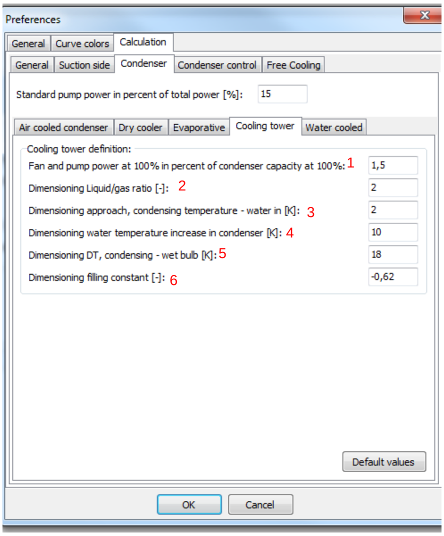

1 : Fan and pump power at 100% in per cent of condenser capacity at 100%

2: LG relation = Mwater/Mair

3: DT=T_c-T_wi

4: DT_water=T_wi-T_wo

5: DT=T_c-T_wb

6: M filling constant

Temperatures dry cooler, PackCalc

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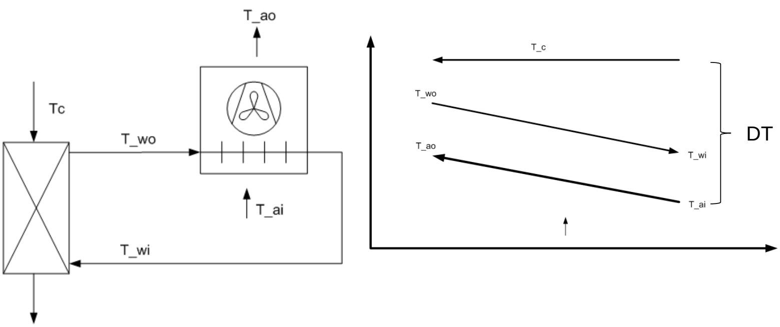

DT= T_c-T_air

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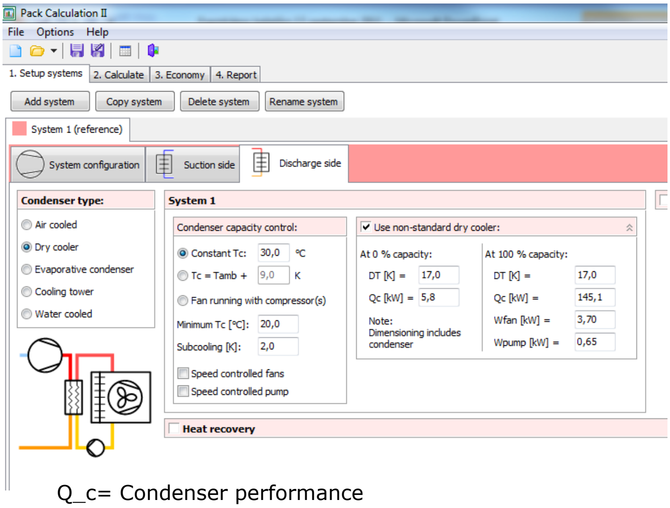

Q_c= Condenser performance
W_fan+W_pump=3% af Q_c=0.03*145.1=4.35
W_pump=15% af 4.35 kW=0.65 kW
W_fan=4.35-0.65= 3.7 kW

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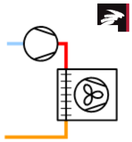
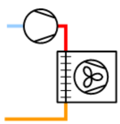
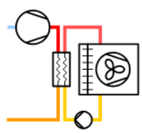
- Air-cooled condenser with constant condensation temperature of 35°C for the entire year



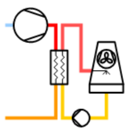
- Air-cooled condenser, where the condensation temperature follows the ambient temperature +9 °C

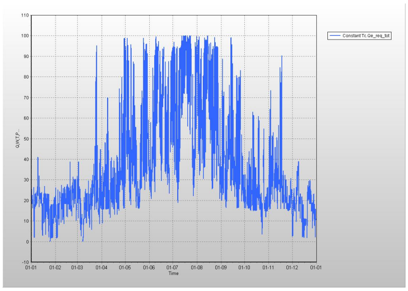


- Air-cooled condenser that emits heat in an air-cooled dry cooler. DT=T_c-T_air=9°K



- Water-cooled condenser that emits heat in a cooling tower. DT=T_c-T_wb=9°K





Load profile

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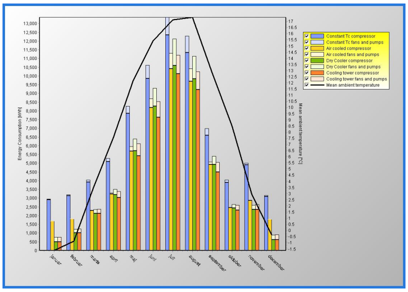
Hungary and load profile for industrial comfort

Energy consumption



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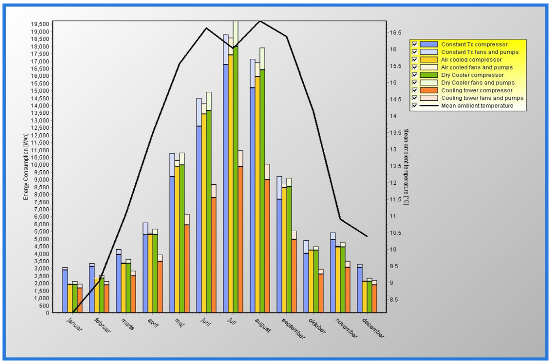

Saudi Arabia and load profile for industrial comfort

Copenhagen, cooling profile: Low energy house

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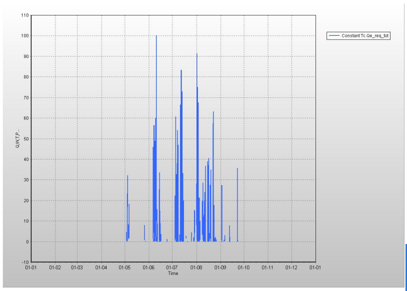

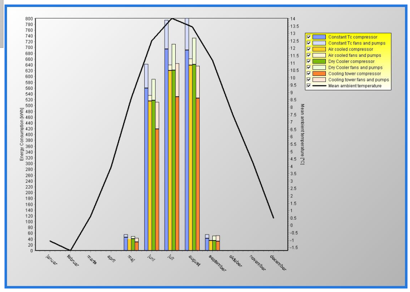

Copenhagen, constant cooling profile =100 kW

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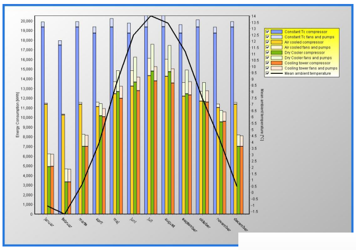

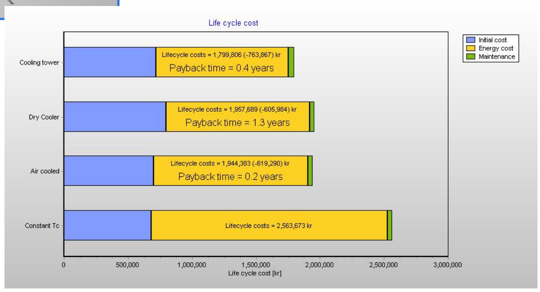

Conclusion

DANISH TECHNOLOGICAL INSTITUTE

- A new cooling tower concept has been developed and tested
- The cooling tower showed good performance
- Improvements can be obtained with better water distribution
- Use nanotechnologies for the surface
- System with rainwater as make up water developed
- Savings with cooling tower compared to dry cooler
- Models implemented in PackCalc
- Good project with all engineering disciplines (Theory and Practice)

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Thank you for your attention

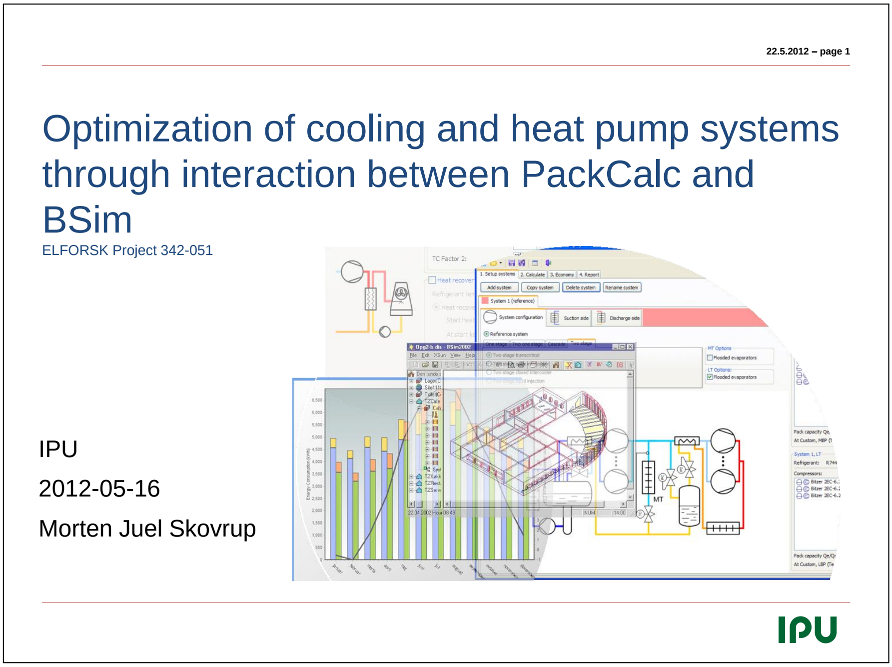
Any questions?

Go back to the table of contents ⩞ or to the timetable ▲

3.12 Optimization Through Interaction Between PackCalc and BSim

Morten Juel Skovrup *(mjs@ipu.dk)*
IPU Refrigeration and Energy Technology

Timetable ▲
Table of contents ⊼



PackCalc and BSim

22.5.2012 – page 2

- Agenda
  - Background
  - Project workgroups
  - Details
    - Auto-generated systems/compressors
    - Auto-selected HX
    - Heat recovery
    - Free cooling
    - Groundwater cooling
  - Next step

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Project goal

22.5.2012 – page 3

- Develop a PC-tool that enables the user to (easily) show the energy-economic consequences of choices of:
  - System layout
  - Components
  - Control strategy
- In air-conditioning and heat pump systems and relating it to:
  - Building comfort
  - Indoor climate
- In different types of buildings…

IPU

Background

22.5.2012 – page 4

- Combining two existing tools:
- BSim
  - Building simulation program
    - Synchronous simulation of moisture and energy transport in constructions and spaces
    - Calculation of natural ventilation
    - Dynamic simulation
- PackCalc
  - Yearly energy consumption of refrigeration systems and heat pumps
    - Static simulation
    - Details later…

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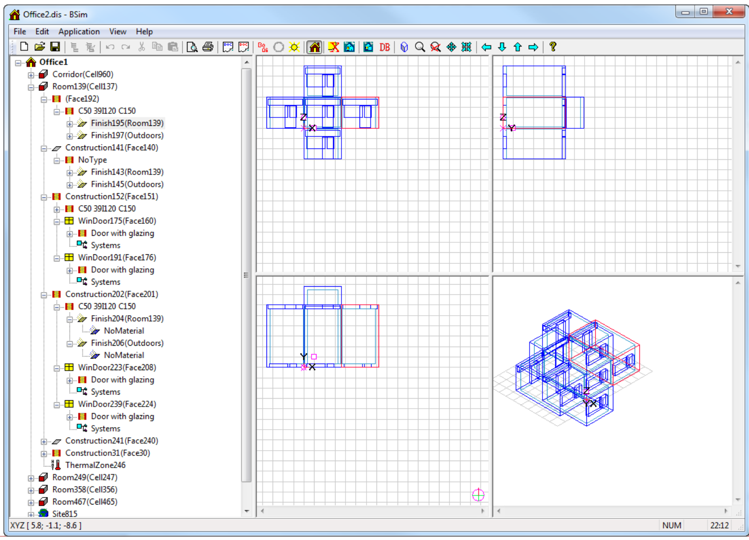
BSim

22.5.2012 – page 5



IPU

PackCalc Basics

22.5.2012 – page 6

- Yearly energy consumption of refrigeration systems
- Calculates every hour throughout the year – i.e. 8760 calculations
- 12 different cycles
  - 3 transcritical cycles
  - 1 heat pump
- Each cycle can be modified by optional
  - Internal heat exchanger
  - Flooded evaporators
  - Secondary circuit on the evaporator side

- 5 different condenser configurations
  - Air cooled, dry cooler, evaporative, cooling tower, water cooled
- Free cooling and heat recovery is available (not at the same time though…)
- 4130 models of commercially available compressor included
- Climate data for 707 cities

- Economics (payback), LCC, TEWI

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Background

22.5.2012 – page 7

- Participants

Project Group

The following companies have generously supported the project by donating data for compressors:

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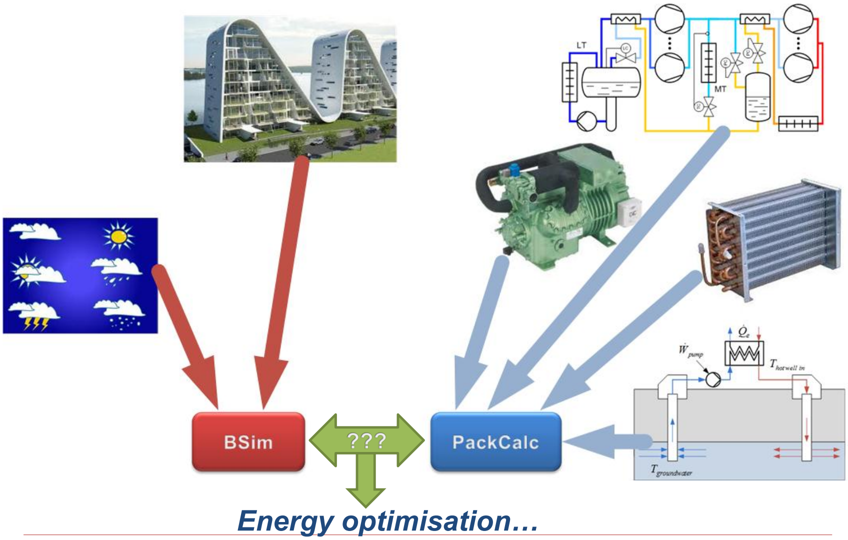
Background

22.5.2012 – page 8



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Workgroups

22.5.2012 – page 9

Data exchange
- BSim → Load profile → PackCalc

Integration
- BSim ↔ PackCalc

Product Improvements
- PackCalc: Groundwater cooling, Free cooling, Heat pump, AC
- BSim: Including technical systems

Profiles
- Use BSim to generate "standard" load profiles (heating and cooling) for use in PackCalc

Case
- Test on users

Smart grid
- Change set points in BSim from external software

IPU

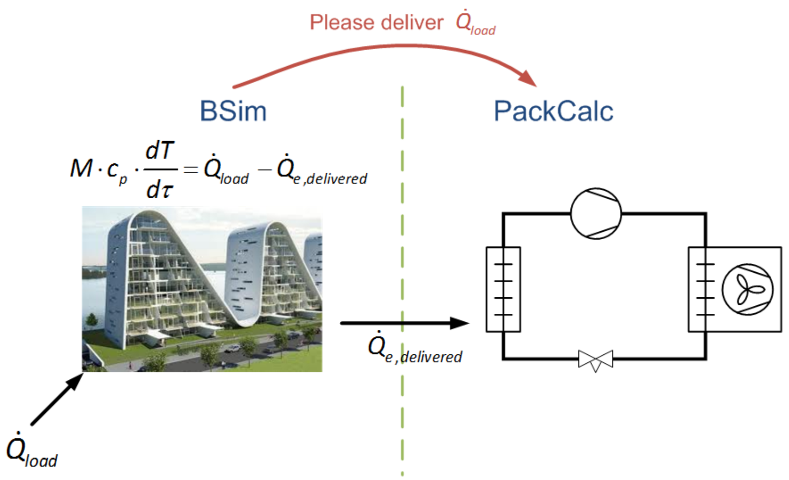
Integration of BSim and PackCalc

22.5.2012 – page 10

- Supported work-flow:
  1. User defines building (construction, orientation, location) in BSim
     - This implies defining "thermal zones" – a thermal zone is one or more rooms that have common indoor climate control
  2. Each of the thermal zones will have a dimensioning cooling requirement (or heating requirement)
  3. The user can select central cooling and/or heating systems. A central system can cover several thermal zones
  4. From the dimensioning requirement (sum of requirement in thermal zones), the user selects a simplified refrigeration system, including options for:
     - Compressors
     - Heat exchangers
     - Free cooling
     - Heat recovery
  5. The user runs a simulation. If the AC system does not supply required capacity, then the indoor climate will change!
  6. Power consumption of AC system is calculated
  7. BSim user can "play" with options
  8. BSim user exports PackCalc model AND load requirement profile so that refrigeration expert can fine-tune the system (real compressor models, HX, …)

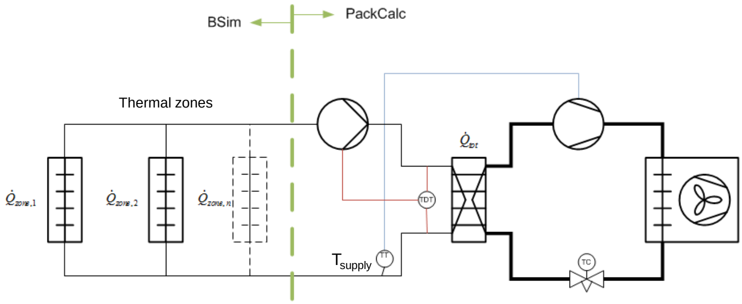
Integration

22.5.2012 – page 11



System

22.5.2012 – page 12



- One-stage cycle
- One-to-one system
- Secondary system “disregarded” – BSim distributes cooling from refrigeration system
- Compressor capacity controlled by supply temperature

$$\dot{Q}_e = UA_{evaporator} \cdot \left(T_{supply} - T_e\right)$$

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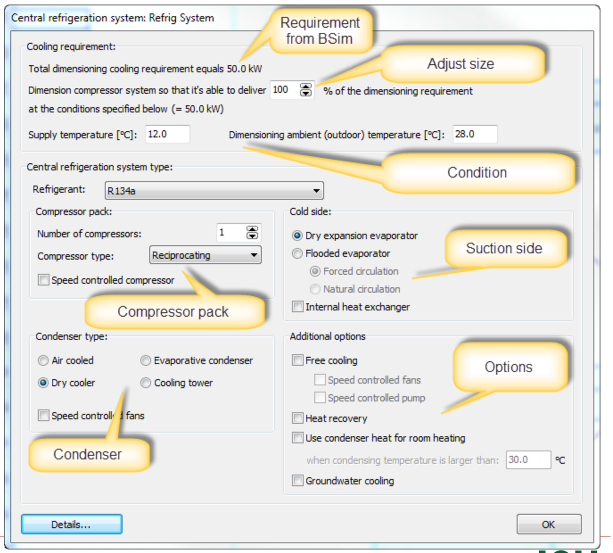
Integration of BSim and PackCalc

22.5.2012 – page 13

- Simplified system:



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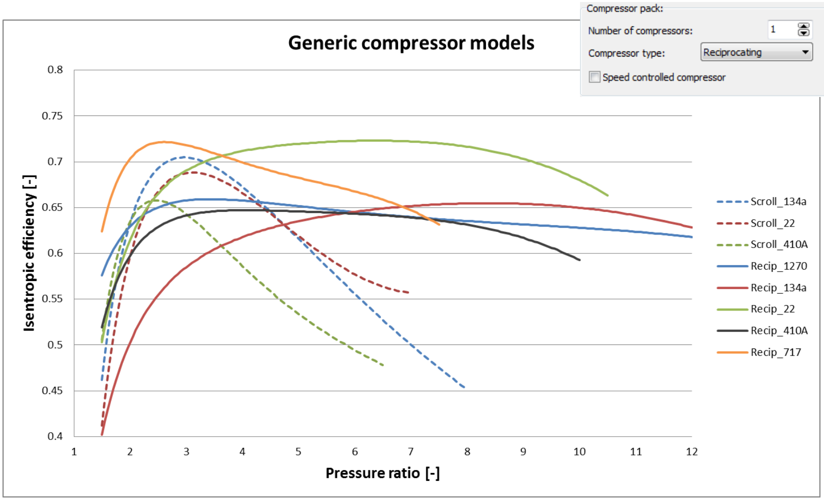
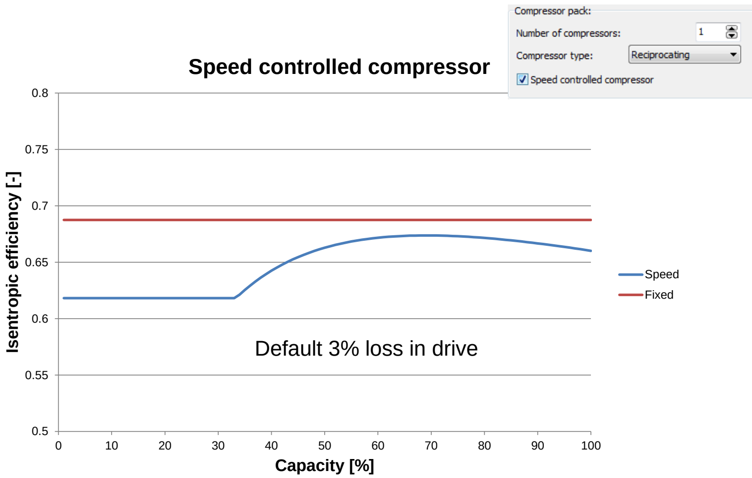
Compressors

22.5.2012 – page 14



Speed control

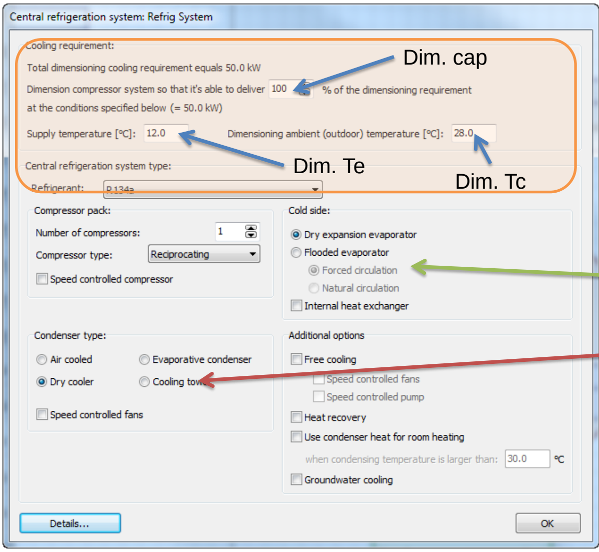
22.5.2012 – page 15



Only for one operating condition (Te,Tc) shape is different at other conditions

Condenser and evaporator

22.5.2012 – page 16



- Using requirement and condition, dimensioning capacities can be calculated (cycle calculation).
- Then using default value for dimensioning temperature differences:

$$UA_{evaporator} = \frac{\dot{Q}_{e,\dim}}{\Delta T_{e,\dim}}$$

$$UA_{condenser} = \frac{\dot{Q}_{c,\dim}}{\Delta T_{c,\dim}}$$

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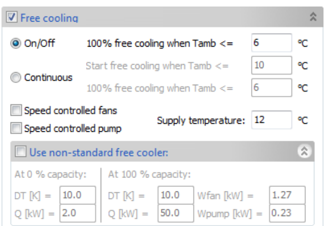
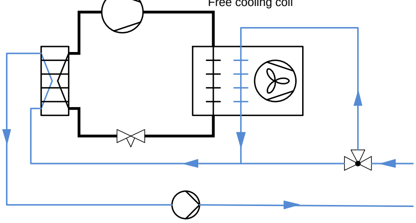
Free cooling

22.5.2012 – page 17

- 2 options:
  - 1) Start free cooling at a certain ambient temperature and then run 100% (default)
  - 2) Start free cooling – typically at a higher – ambient temperature and ramp up to 100% at another temperature





- To calculate the power consumption of the fans and the pumps, you have to specify a heat exchanger (like specifying an air cooled condenser):

$$\dot{Q}_{freecool} = UA_{freecool} \cdot \left(T_{supply} - T_{amb}\right)$$

IPU

Heat recovery

22.5.2012 – page 18

☑ Heat recovery

Refrigerant temp. out of recovery heat exchanger: 24 °C

◉ Heat recovery is a function of ambient temperature:

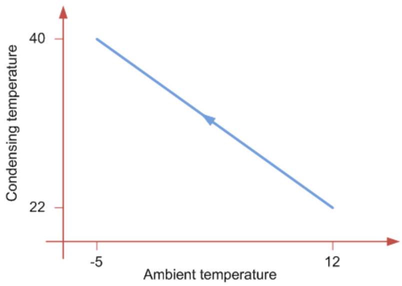
- Start heat recovery when T_amb = 12 °C
- At start keep high pressure at: 22 °C
- Maximum heat recovery when T_amb = -5 °C
- High pressure at maximum heat recovery: 40 °C

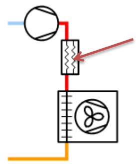
○ Run heat recovery at all times:

- Keep high pressure at: 40 °C

Monthly schedule (select months where heat recovery is enabled):

☑ January	☑ February	☑ March	Q1
☑ April	☑ May	☑ June	Q2
☑ July	☑ August	☑ September	Q3
☑ October	☑ November	☑ December	Q4





Heat recovery heat exchanger

The heat recovery function is overriding the condenser control if high pressure is below the specified heat recovery pressures…

IPU

Groundwater cooling

22.5.2012 – page 19

- Cooling:

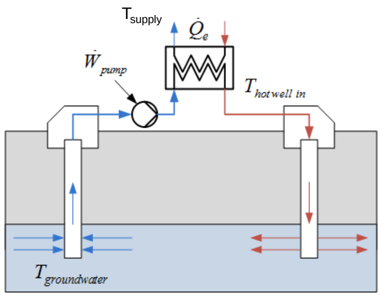
Cooling:

- Dimensioning cooling capacity [kW]: 50
- Temperature of water into hot well [°C]: 15

Common:

- Total pressure drop in piping [bar]: 2
- Pump efficiency [-]: 0.5
- ☑ Custom groundwater temperature:
- Groundwater temperature [°C]: 7

From BSim



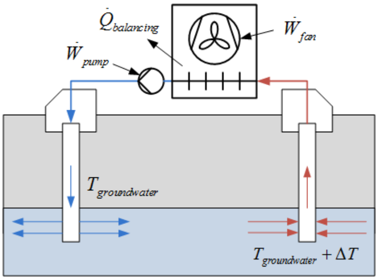
- Straight forward calculation:
  - Provides cooling up to dimensioning capacity (if more required, then mechanical cooling sets in)
  - Power consumption = W_pump
  - Keep track of heat added to hot well!

IPU

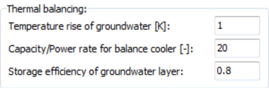
Groundwater cooling

22.5.2012 – page 20

- Balancing
  - Full balancing is always required!
  - Balancing is calculated at the end of the simulation:



- Required balancing is based on storage efficiency and temperature rise of ground water:



$$\dot{Q}_{balancing} = \eta_{storage} \cdot \dot{Q}_{cooling}$$

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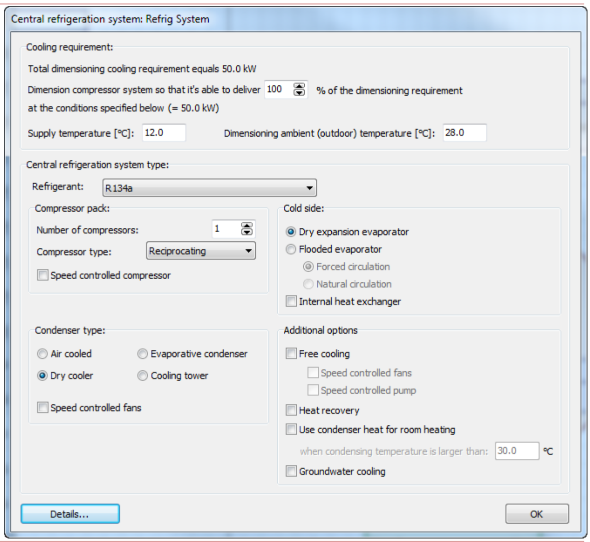
Sum up

22.5.2012 – page 21

- A lot of default values based on a few clicks in:
- Therefore options should be reviewed by refrigeration expert… BSim interface is for testing possibilities.



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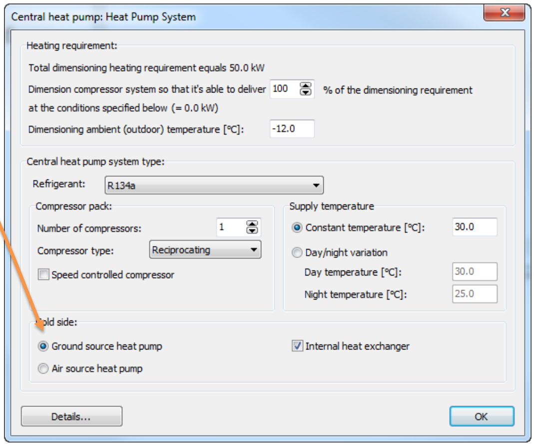
Heat pump

22.5.2012 – page 22

- More or less the same as for refrigeration system:
- Except
- Only difference between ground and air source is:
  - Evaporator ambient temperature (which temperature to use from profile)
  - Default dimensioning temperature difference when calculating UA of evaporator…



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Usage

22.5.2012 – page 23

- So why all of this?
- Where's the results?
- Currently last hands are put on BSim, so that the interface is fully supported
  - Challenge is to go from "calculation of requirement" to calculation of "what happens if requirement is not met?"
- Final testing begins beginning next week…

IPU

Conclusion

22.5.2012 – page 24

- Connecting BSim and PackCalc enables BSim engineers to quickly quantify consequences of:
  - Changing operating conditions for AC/HP system (changing supply temperature)
  - Changing control strategies for AC system
  - Testing:
    - Free cooling
    - Groundwater cooling
    - Speed control
    - …

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Next step

22.5.2012 – page 25

- BSim will (as part of this project) be prepared to accept set points from other software…

- This means that scenarios like the following can be investigated in the near future:
  - What is the consequences of cooling a building in the night and turning of the AC during daytime?
    - Economic
    - Comfort
  - What is the consequences of turning off cooling/heating for a short time?
  - …

- This can be done using realistic building models and realistic models of AC and HP systems

IPU

Next step – PackCalc

22.5.2012 – page 26

- Integration with other energy systems
  - Heat pump on top of refrigeration system
  - Compared to gas, oil, …

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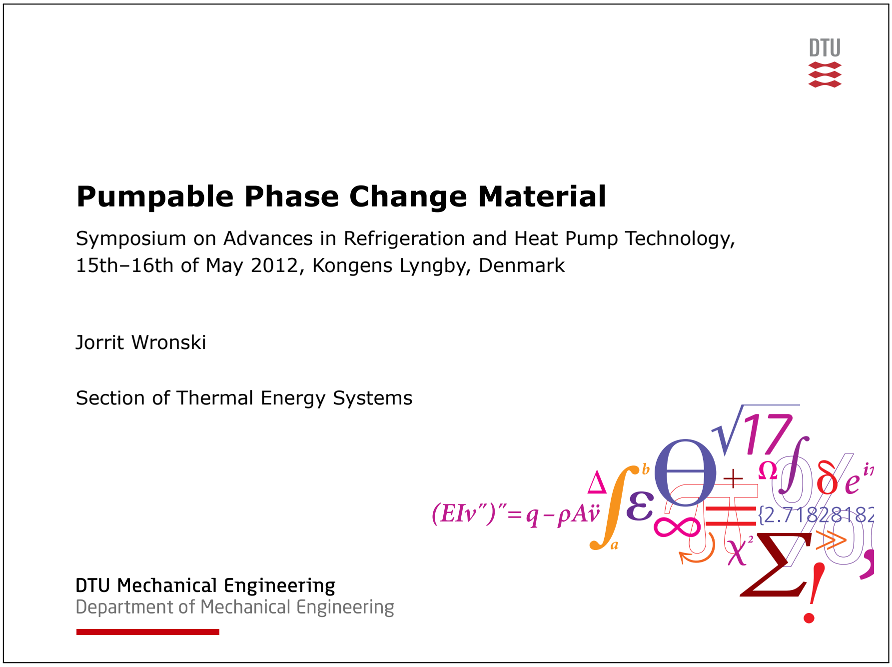
Go back to the table of contents ⯒ or to the timetable ▲

3.13 Pumpable Phase Change Material

Jorrit Wronski *(jowr@mek.dtu.dk)*
DTU Mechanical Engineering

Timetable ▲
Table of contents ⏶



DTU

Contents

What is a PCS?

Goal and Partners

Production

Experiments

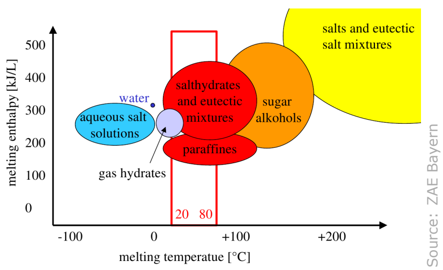
Conclusions

2 **DTU Mechanical Engineering, Technical University of Denmark** Pumpable Phase Change Material 15.5.2012

DTU

Introduction to PCM

- Phase change material
- Cyclic phase shift
- High energy density
- Gas and salt hydrates
- Paraffin and fatty acids
- Additive or in storage vessels



3 **DTU Mechanical Engineering, Technical University of Denmark** Pumpable Phase Change Material 15.5.2012

DTU

Introduction to PCM

- Phase change material
- Cyclic phase shift
- High energy density
- **Gas and salt hydrates**
- Paraffin and fatty acids
- Additive or in storage vessels

Source: Wikipedia

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DTU

Introduction to PCM

- Phase change material
- Cyclic phase shift
- High energy density
- Gas and salt hydrates
- **Paraffin and fatty acids**
- Additive or in storage vessels

Source: Rubitherm GmbH

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DTU

Introduction to PCM

- Phase change material
- Cyclic phase shift
- High energy density
- Gas and salt hydrates
- Paraffin and fatty acids
- **Additive or in storage vessels**

Source: Fraunhofer ISE

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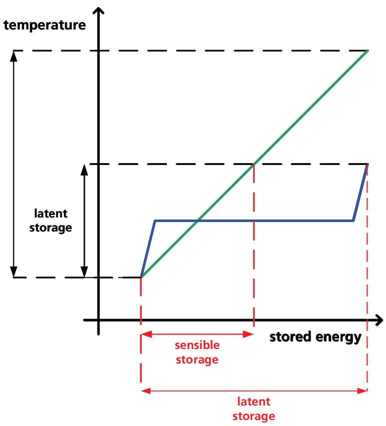
What is PCS?

- Phase change slurry
- Dispersion:
  suspension (solid in liquid)
  or
  emulsion (liquid in liquid)
- Heat transfer fluid
- Thermal energy storage



Source: Fraunhofer UMSICHT

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One Substance PCS

- Coexisting phases of one substance
- Homogeneous if discharged
- Regularly formation
- Limited temperature range

Source: Fraunhofer UMSICHT

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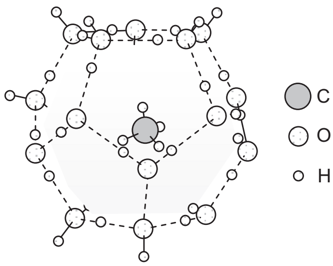
Hydrate Slurries

- Gas (or salt) hydrates in aqueous solution
- Non-mechancal formation
  $n \cdot H_2O + M \xrightarrow{\Delta h_{fus}} M(H_2O)_n$
- Pressures >10 bar for gas



Source: [1]

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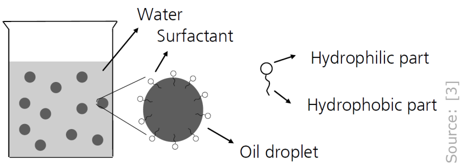
Multi Component PCS

- Liquid continuous/carrier phase and dispersed phase change material
- Stabilised by shell or surfactant
- Sensitive to pumping
- Could include hydrates [2]



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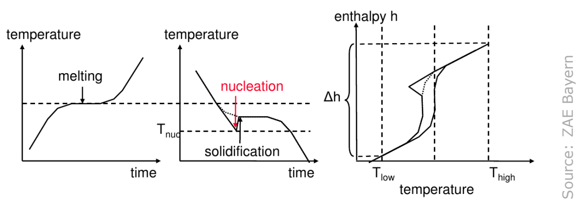
Motivation

- Subcooling for charging [4]
- Charge state, capacity [5]
- Analysis of phase change [5]
- Difficult control

⇒ Hinders application



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Project Partners

- Funded by German Federal Ministry of Economics and Technology (BMWi), reference 032747B (2009–2013)
- "Paraffin-water Emulsions for applications in building supply systems"
  Original German title: "Emulsionen aus Paraffinen und Wasser für Anwendungen in Versorgungssystemen der Gebäudetechnik"

Fraunhofer UMSICHT

- Develops PCS
- Rheology and heat transfer measurements
- **Charge state sensor**

E.ON Energy Research Center
EBC | Institute for Energy Efficient Buildings and Indoor Climate

- Pipe network experiments
- Multiphase flow simulations

JÜLICH FORSCHUNGSZENTRUM

- NMR experts
- Lab facilities
- No project partner, supported charge state sensor

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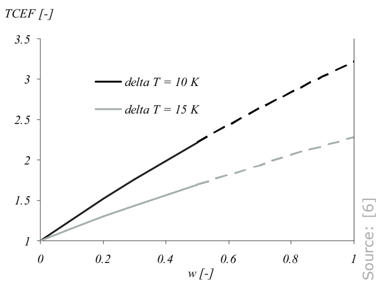
The PCS

- Paraffin-in-water dispersion
- 30 % PCM
- Nucleating agent
- Surface active agent
- Carrier fluid water



10 **DTU Mechanical Engineering, Technical University of Denmark** Pumpable Phase Change Material 15.5.2012

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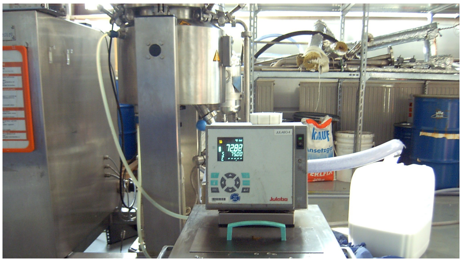
Production I

- Preheat vessel to sufficiently high temperature
- Prepare machines and ingredients



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Production II

- Warm water in emulsifier
- Melt paraffin separately

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Production III

- Add molten PCM to water
- Stir gently and let temperature equalise

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Production IV

- Warm nucleating agent until melts
- Blend surfactant in

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Production V

- Add to water and paraffin
- Emulsify until desired droplet size distribution is reached

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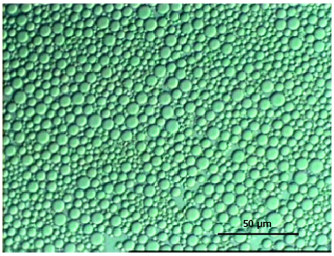
Outer Appearance

Source: [6]

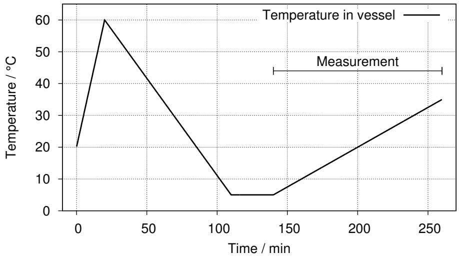


Source: [7]

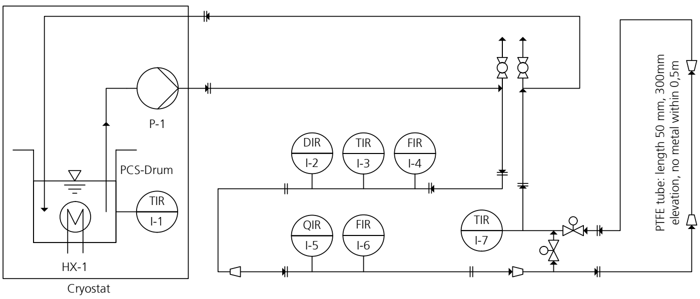
16 **DTU Mechanical Engineering, Technical University of Denmark** Pumpable Phase Change Material 15.5.2012

Approach

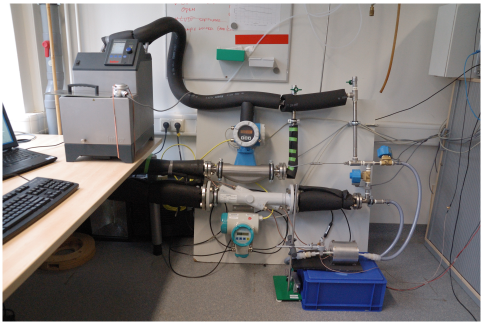
- Monitoring properties during a test cycle:
  Nuclear magnetic resonance (NMR) detects state [8]
  Paraffin density changes by 10 % [9, 10]



Test Rig I



Test Rig II



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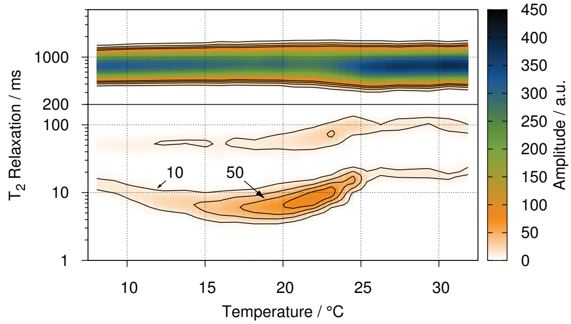
NMR Measurement I



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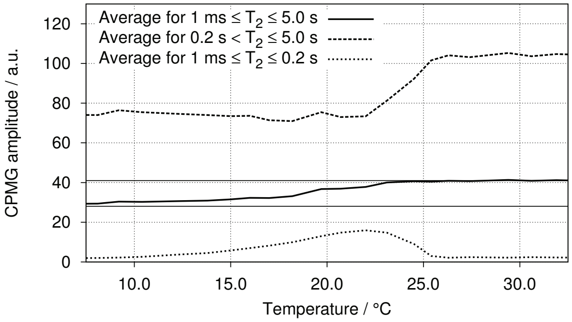
NMR Measurement II



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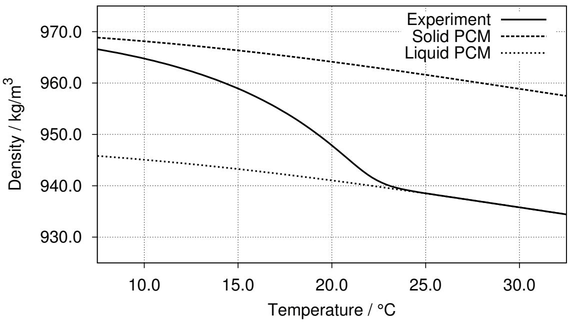
Density Measurement



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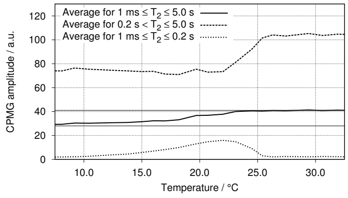
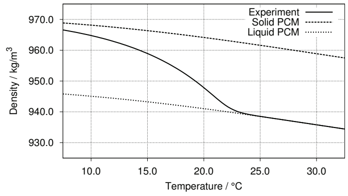
Comparison





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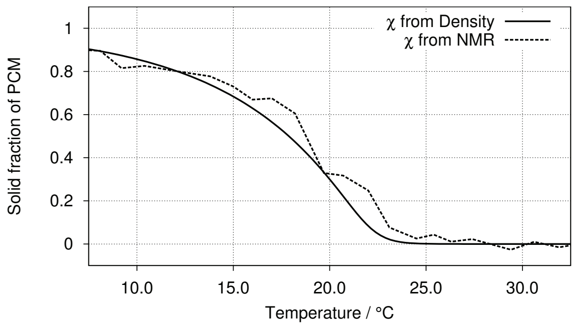
Solid Fraction



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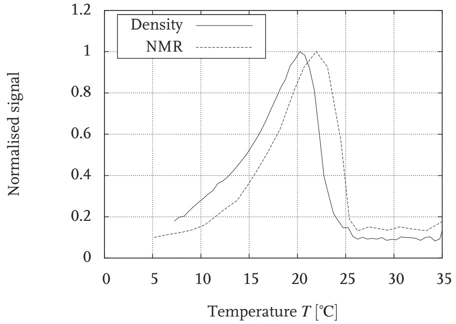
Melting Material



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Concluding Remarks

- Phase change observed
- No absolute values, NMR calibration
- Hysteresis in density measurements
- Cooling not studied

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